Small Scale and Micro Combined Heat and Power

Martin Alan Smith

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Use of heat from generation station.

50. – (I) It shall be the duty of the Central Authority to investigate methods by which heat obtained from or in connection with the generation of electricity may be used for the heating of buildings in neighbouring localities, or for any other useful purpose, and the Authority may accordingly conduct, or assist others in conducting, research into any matters relating to such methods of using heat.

Extract from the *Electricity Act,* 1947 – PART IV, section 50, clause I.

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Abstract

This thesis has concluded that conventional CHP is inappropriate for single dwelling domestic application. The incorporation of a vapour compression heat pump was assessed for domestic application and initial modelling indicated the potential of CHP/HP.

The development of the prototype plant, largely from commercially available components, demonstrated the practical viability of domestic co-generation. Analysis of experimental results demonstrated that CHP/HP addressed the problems associated with conventional domestic CHP. The first law performance of CHP/HP operation was significantly improved over that of CHP. Second law exergy analysis favoured CHP operation as a consequence of the higher thermal and lower electrical deliveries experienced in CHP/HP operation.

A validated computer model was developed to extend the experimental results. Analysis of simulated results shows that CHP/HP operation can extend the envelope of economic operation of domestic co-generation with respect to fuel costs. Analysis also highlighted the importance of low maintenance costs, particularly for CHP/HP operation.

The heat pump incorporation enhances the environmental performance of domestic co-generation plant, allowing for a greater displacement of emissions than for a conventional plant. In circumstances where no economic advantage of heat pump incorporation is apparent, CHP/HP operation significantly reduces emissions compared to CHP operation. Additionally, when economic factors are such as to marginalise any modes of operation, heat pump incorporation continues to displace a significant proportion of conventional energy supply and hence emissions.

Future research will examine the potential of fuel cell-based CHP/HP systems and should concentrate on heat pump design and appropriate control systems.

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Nomenclature

A	model constant	
B	model constant	
C	model or thermal constant or annual cost	£
COP	heat pump coefficient of performance	
D	model constant or energy demand	kW or kWh
D'	adjusted energy demand	kW or kWh
E	exergy	kW or W
I	irreversible losses or current	kW, kJ or A
L	plant load factor	
N	installed grid capacity of a generator plant	
P	pressure or mass of emissions evolved	Pa or kg
Pr	Prandlt number	
Q	heat or energy transfer	kW or W
R	Gas constant or model constant	
Re	Reynolds number	
S	specific emissions of a fuel	kg/kWh
T	Temperature	K or ${}^0C$
U	U Value	kW/m
V	volume or voltage	m^3 or V
a	area	m^2
c	unit cost	$£/kWh$ or $£/kJ$
cp	specific heat capacity at constant pressure	kJ/kgK
d	external diameter of engine block	m
g	gains	kW or W
h	heat transfer coefficient or specific enthalpy	kW/m^2K or kJ/kg
j	heat pump COP exponential constant	
k	thermal conductivity	kW/mK
m	mass flow	kg/s
$\dot{m}$	mass or mass flow	kg or kg/s
n	thermal geometric constant; physical/time node	
p	specific emissions	kg/kWh
q	energy	kJ or J
r	discount rate	
savings	financial saving	£, £/day or £/year
savings-p	financial savings under steady state condition	£, £/day or £/year
w	work or electrical output	kW
x	species or constituent of a mixed flow	

Δ	parameter uncertainty	
α	absorptivity of glass	
β	saving to cost ratio for cogeneration economics or regression constant	
ε	specific exergy or heat exchanger effectiveness	*kJ/kg*
η	first law efficiency	
λ	engine (or CHP plant) heat to power ratio	
μ	dynamic viscosity	*kg/ms*
ρ	density	kg/m^3
σ	Stefan-Boltzman constant	kW/m^2K
τ	model time step size	*s*
ψ	second law efficiency	

Subscripts

1 to 9	points within prototype plant
air	air or air flow
boiler	boiler
building	general building environment
bulk	EHE bulk water temperature
b	engine heat/power ratio constant
c	engine heat to power function constant
ccgt	refers to a combined cycle gas turbine generator
chp	Combined Heat and Power Plant
chphp	Combined Heat and Power/Heat Pump Plant
coal	refers to a coil fired Rankine cycle generator
conv	connective heat transfer or losses
COP	refers to heat pump COP
COP1	refers to an experimentally derived heat pump COP
COP2	refers to a derived by-pass heat pump COP
COP-i	refers to a minimum operational tolerable part load COP characteristic
COP-ii	refers to a minimum financial/environmental tolerable part load COP characteristic
e	electrical
ehe	exhaust Heat Exchanger
ehe-in	EHE input parameter
ehe-p	potential steady state EHE parameter
eng	engine thermal output or losses
env	environmental acquisition
ex	refers to exhaust gas thermal availability or losses
f	fuel
gen	generator parameter
hp	Heat Pump
hp-in	heat pump heat exchanger input
hploss	heat pump losses

hp-p	potential steady state heat pump parameter
i	state or mode under consideration
in	input
loss	losses
Lt	light
m	maintenance
max	refers to a maximum value of a parameter
min	refers to a minimum value of a parameter
n	mode within a system or a time period
ng	refers to natural gas (approximated to methane)
o	initial or reference state
oil	refers to an oil fired Rankine cycle generator
old	value of parameter from previous model iteration
out	output
pr	plant room
r	refers to plant running costs
rad	radiative heat transfer or losses
rejected	thermodynamically rejected energy.
s	surplus electrical availability
surface	plant or component surface
th	general or total plant thermal parameter
th-p	potential steady state plant thermal parameter
wt	water or LPW system parameter

Abbreviations

PBP	Pay Back Period
CHP	Combined Heat and Power
CHP/HP	Combined Heat and Power incorporating a Heat Pump
EHE	Exhaust Heat Exchanger
LPW	Low Pressure Water
NPV	Net Present Value

1 Introduction

Combined heat and power (CHP) or co-generation - the concurrent delivery of electrical and thermal energy from a single plant - is an established technology in industrial and commercial applications. The use of co-generation has considerable economic advantages over conventional forms of energy supply, due to high fuel utilisation. Additionally, co-generation significantly reduces the environmental effects of fossil fuel consumption. Co-generation technology is simple and commercially available and as such has been cited as an intermediate method of reducing carbon dioxide emissions from fossil fuels: recent government policies have sought to encourage the widespread application of CHP plants to this end.

To maximise the environmental advantages of co-generation, it is necessary to implement the installation of CHP in the largest energy market – the residential sector. Many industrial sites are well suited to co-generation applications, as are dense residential areas. However, the widespread implementation of co-generation in the UK residential sector is complicated by the suburban character of British towns and cities. As the UK urban population is housed mainly in small individual low-rise dwellings, the supply of heat from CHP (in the form of steam or hot water) would be expensive and complex. It would involve the installation of an extensive heat distribution system. In contrast, the distribution of thermal energy to dense multiple occupancy housing, as found in Eastern Europe, is relatively simple and cost effective. This type of supply has been viewed as inappropriate and prohibitively expensive for UK suburban application.

This thesis presents a solution that may allow for the widespread cost effective implementation of co-generation to the UK residential sector. In contrast to the use of one large plant in supplying many small dwellings, it is proposed that a small single CHP plant incorporating a heat pump would supply an individual dwelling. Such a plant overcomes many of the problems associated with UK residential co-generation. The following introduction will define the aims and describe the organisation of this thesis. Key terminology will also be introduced.

1.1 Aims

This thesis aims to demonstrate theoretically and practically that the concept of heat pump inclusion in a domestic co-generation plant is valid. A salient aim of the thesis is to compare the proposed domestic co-generation plant incorporating a heat pump with one without, in both economic and environmental terms. An additional aim is to assess the practical feasibility of domestic co-generation, with the construction of a prototype plant, which would largely utilise commercially available components.

1.2 Thesis Structure

A review of co-generation will be carried out in Chapter 2, which will examine previous research undertaken in the field of domestic co-generation. Current CHP technologies and practices will also be reviewed, along with the historical and institutional aspects of UK co-generation. The review of co-generation will be followed by a case study of an operational small scale CHP plant (installed in The Queens Building, DeMontfort University), which is presented in Chapter 3. The case study is intended to be an extension of the review. It will introduce analytical methods that will be employed in later chapters.

Chapter 4 formally introduces the concept of heat pump (HP) incorporation into a CHP plant – to form a CHP/HP plant. This will be accompanied by the results and subsequent analysis of preliminary modelling, to assess the CHP/HP concept. This modelling was carried out to examine the CHP/HP concept before any investment was made in the prototype plant.

The development of the laboratory prototype plant is chronicled in Chapter 5. The sources of components are identified and plant development is discussed. Supporting design information is presented in Chapter 5 and in the appendix. The design of the associated test rig and instrumentation is also detailed.

Experimental results obtained from the prototype plant are presented, analysed and discussed in Chapter 6. Transient performance characteristics of the prototype plant are considered in Chapters 6 and 7. First law and second law thermodynamic analysis is carried out for representative steady state conditions, and is subsequently discussed, in Chapter 6. Transducer characteristics and actual experimental data pertaining to the analysis are given in the Appendices D and E.

Experimental results were subsequently utilised in the construction of a concept evaluation model. The development of the concept evaluation model is detailed in Chapter 7. The aim of the concept evaluation model was to simulate the prototype plant under various economic and operational conditions that could not be achieved in the laboratory environment. The validation exercise carried out for the concept evaluation model is also presented in Chapter 7.

Chapter 8 utilises the validated concept evaluation model to assess the viability of domestic co-generation, both with and without heat pump incorporation. Limits of economic operation are established for both types of co-generation. Comparative analysis is carried out to assess the CHP/HP concept. Conclusions are drawn in Chapter 9, making reference to practical and simulated results and analysis.

1.3 Terminology

Specific terms used in the thesis are defined in this section for future reference:

Conventional energy supply is defined as electricity supplied by the utility grid and boiler-supplied heat. *Conventional CHP* denotes a CHP plant that does not include a heat pump. The term *co-generation* is used when both conventional CHP and CHP/HP plants are discussed collectively. *Environmental performance* considers the percentage reduction in global carbon dioxide emissions due to co-generation operation. Other terms are defined in relevant sections.

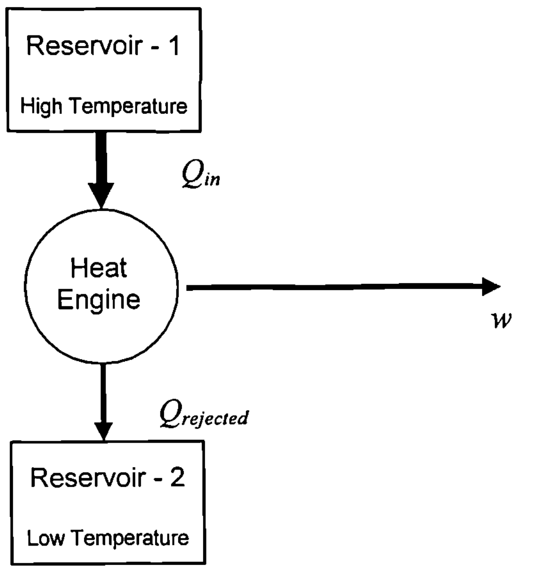
2. Review of Combined Heat and Power

2.1 Introduction

The following chapter is a review of combined heat and power (CHP) plants, practices and other material relevant to the thesis. The thermodynamic basis and general economics will be covered initially, followed by a review of large scale (above 0.5MWe) and small scale (below 500kWe) plant. The market development of CHP within the UK is summarised. Domestic scale CHP and relevant application of heat pumps will be reviewed. Many of the points made in this chapter are demonstrated in the case study that constitutes Chapter 3.

2.2 The Thermodynamic Basis of CHP

All thermal generators currently used can be classed as heat engines. This includes steam turbine plant, internal combustion engines and Stirling engines, which drive generators. Due to the second law of thermodynamics, some heat must be rejected from a heat engine when converting heat to work. With reference to Figure 2.1, the heat engine, which is operating between two thermal reservoirs, consumes Q_{in} units of heat and produces w units of work. In doing so $Q_{rejected}$ is rejected into the low temperature reservoir.



$$\eta = \frac{w}{Q_{in}}$$

Figure 2.1 Heat Engine

In practice, the fuel conversion efficiency of most heat engines is in the region of 20% to 40%, depending on the type of thermodynamic cycle used. See Table 2.1 below for a summary of common thermodynamic cycle efficiencies.

Table 2.1 Work/ Electrical Conversion Efficiencies [1]

Cycle	**Conversion Efficiency** η **(%)**
Spark Ignition Engine	10 to 25
Stirling Engine	10 to 25
Gas Turbine	<20
Fuel Cell	50 to 70

In most thermal power plants the rejected thermal energy ($Q_{rejected}$) is lost to the environment, via cooling towers or radiators. If the rejected heat is recovered, it may be employed for space heating or process use: this is the principle of CHP or co-generation. By the partial recovery of the rejected heat, the total efficiency of a typical heat engine can be raised from 30% to 70%. This has both economic and environmental implications.

2.2.1 Economic Implications Of CHP

It is often convenient to view a CHP plant as a boiler with a relatively low efficiency that provides electricity at no cost, by assuming that the associated plant running costs are incurred in the production of heat.

Consider a building with a given electrical and thermal demand:

i) Conventional energy demands would be met by boiler plant and utility supplied power from centralised plant. Fuel costs are incurred by the consumption of fuel by the boiler and by the centralised power plants.

ii) In the case of a CHP plant, costs are incurred by the fuelling and maintenance of the plant. If it is assumed that all the costs are incurred by the heat production, the heating costs of the building increase relative to those incurred by a conventional boiler. However, *if electricity demand is satisfied at no extra cost* (under the above assumption) this may off set the higher heat costs and a net saving is achieved.

The exact level of saving depends on many factors such as plant efficiencies, fuel costs and maintenance costs. The economic analysis of CHP plants is developed fully and applied to a case study in Chapter 3.

2.2.2 Environmental Implications of CHP

The implementation of CHP will have environmental benefits due to similar reasons cited in section 2.2.1. In general, less fuel is consumed to satisfy a given electrical and thermal demand (with respect to conventional forms of energy plant), hence less emissions are evolved. The environmental aspects of CHP are analysed further in Chapter 3.

2.3 CHP Plant

The following section will review the different types of plant used in CHP applications. The type of plant used depends primarily on the level of demand that the plant is expected to meet. Figure 2.2 defines the relative demands and identifies the relevant applications.

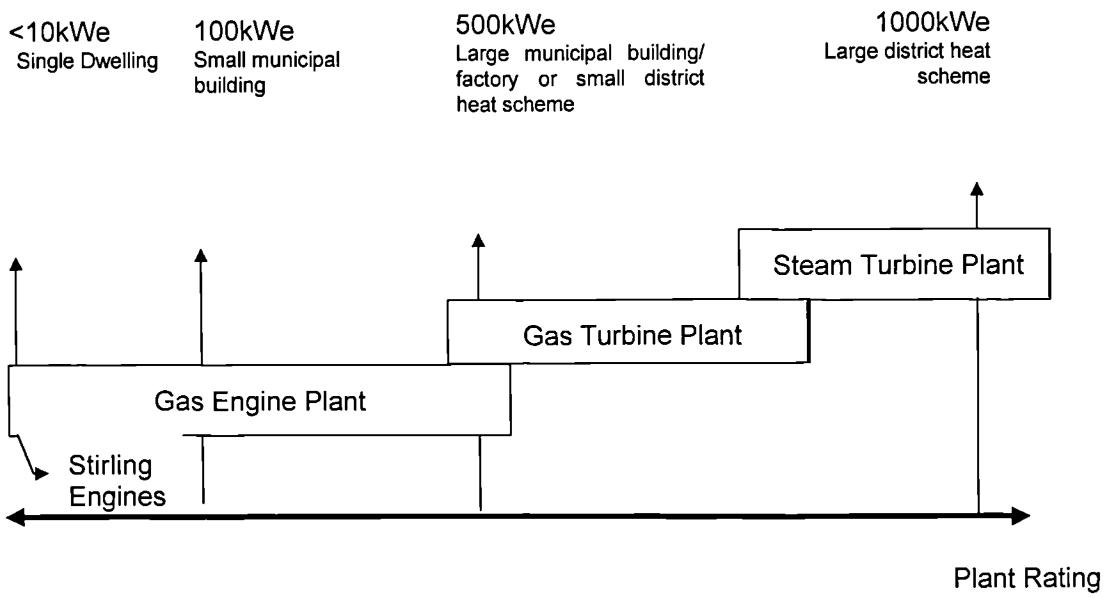


Figure 2.2 CHP Plants and Applications

2.3.1 Large Scale CHP Plant - District Heating or Large Industrial Processes

In this thesis, large scale CHP will be defined as a plant that has an electrical output greater than 500kWe. Typical applications for such large plant are district heating schemes or industrial plants (or both in some instances).

2.3.1.1 Steam Turbine Plant

Steam turbine plant has been used for co-generation since the 1900s [2,3]. Electrical rating of such plant ranges from approximately 200kWe to 200MWe, but is generally rated between 5MWe and 40MWe.

Applications of steam turbine co-generation plant are district heating schemes or industrial processes. Many continental cities partially utilise co-generation steam turbine plants and in particular Eastern European cities have extensive utility city wide heating systems based on steam turbine plant (see Appendix A for a summary of a typical plant in Budapest, Hungary). In general, most district heating schemes burn primary fuel to raise steam and are classed as topping cycles. In previous decades, there was considerable research in this field [4,5] but as will be discussed later, this type of technology has not been prevalent in the UK (see Section 2.4).

The application of steam turbine co-generation technology is particularly well suited to industrial processes which require steam [2,3,6] such as paper mills or chemical processing. Industrial applications may often utilise waste products for fuel or drive the steam cycle from waste heat (bottoming cycles).

Steam turbine plant can be further sub-divided into two groups [7]: back pressure and extract steam turbines. The use of a particular type of plant will depend on the application, as discussed in the following sections.

2.3.1.1.1 Back Pressure Steam Turbine Plant

Back pressure steam turbine (BPST) plant is relatively simple and predominantly found in district heating applications. With reference to Figure 2.3, a BPST plant utilises a modified Rankine cycle, in which steam raised in a boiler drives a non-condensing turbine with a single pressure exhaust. The exhaust steam is passed into a heat exchanger, where a secondary circuit transfers the waste heat to the point of use. The turbine is used to drive a generator or to provide pumping in some industrial applications. BPST systems have a lower mechanical efficiency in comparison with a conventional Rankine cycle plant, as high exhaust temperatures are required to satisfy heating requirements, and hence less energy is available for the turbine to convert into work.

A typical application of a BPST system is the Biker project in Newcastle-upon-Tyne (source: Coppus, USA), which uses a 150kWe radial steam turbine to power and heat 200 flats. The boiler is partially fuelled from domestic refuse incineration.

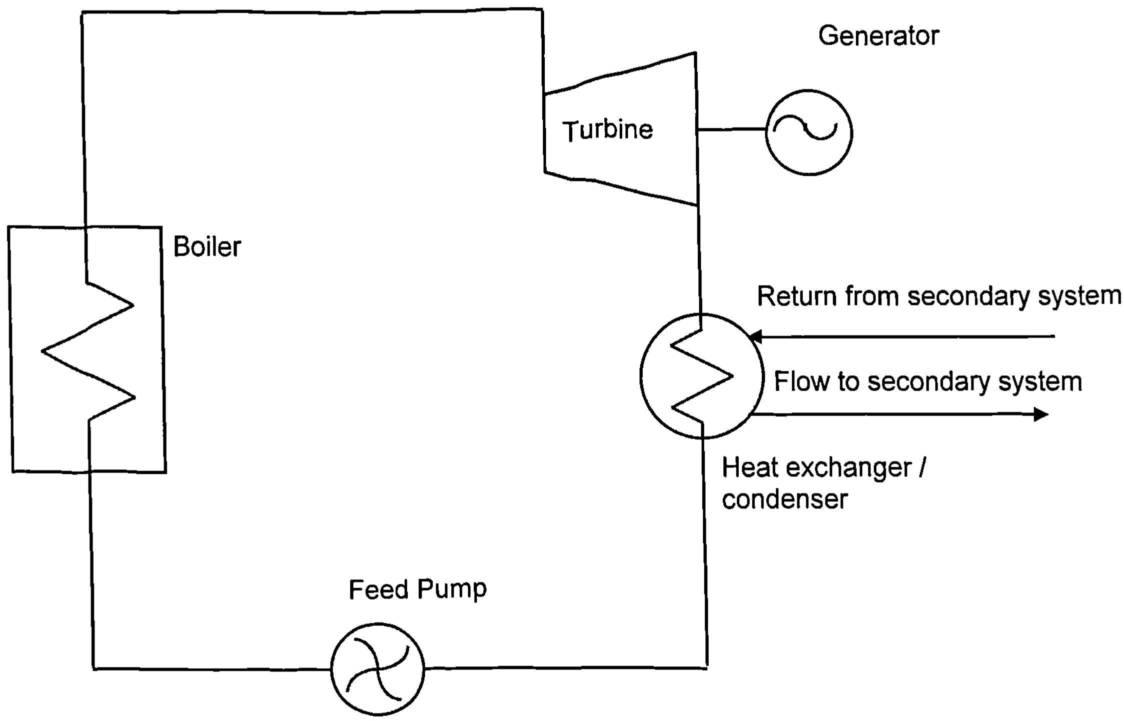


Figure 2.3 Back Pressure Steam Turbine Plant

2.2.1.1.2 Extract (Pass-out) Steam Turbine Plant

Extract steam turbine plants are commonly found in industrial applications, as opposed to district heat applications. With reference to Figure 2.4, steam is raised in a boiler and fed into a multi-stage turbine. Some steam is extracted at an intermediate pressure while the rest passes through to the low pressure turbine stages. The extracted steam is then either directly used in process heating or to transfer heat to a secondary circuit. Both streams are condensed and re-combined to feed the boiler. Extracting steam between turbine stages allows greater control of steam temperature and pressure, which may be critical in an industrial process. The work output of the turbine (and hence conversion efficiency) will depend on how much steam is extracted.

A typical application of an extract steam turbine plant is the Scott Paper Mill, in Alabama, USA [6]. Waste wood chips (from the production process) raise steam in a boiler, which drives a 32MWe extract turbine. Extracted steam is used in the production processes.

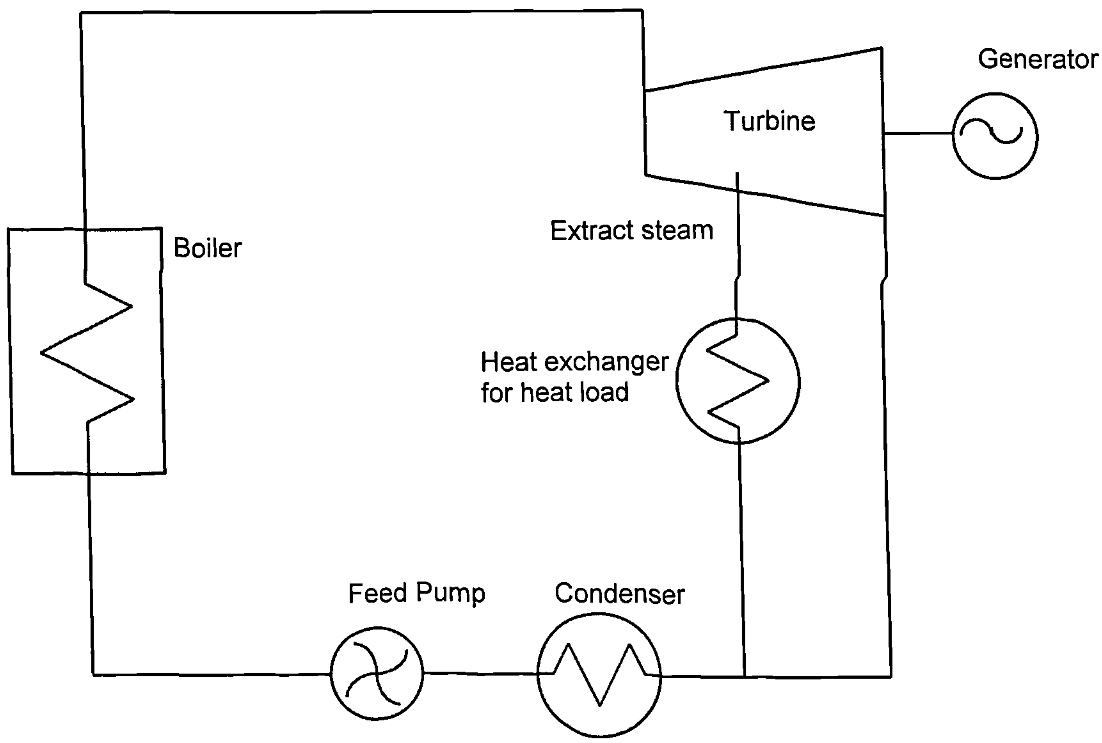


Figure 2.4 Pass-Out Steam Turbine Plant

2.3.1.2 Single Cycle and Combined Cycle Gas Turbine Plant

Gas turbine-based CHP systems have a number of distinct advantages over steam turbine plant [9,10]. Owing to the aeronautical lineage of most designs, gas turbines have the following advantageous characteristics:

- Compact and self contained: no requirement for external ancillary equipment, unlike steam turbine plant.
- High power to weight ratios.
- Comparatively simple and reliable.
- Rapid starting: minutes as opposed to hours for steam turbine plant.
- Comparatively high production and standardised designs.

Figure 2.5 illustrates a simple gas turbine CHP plant. The gas turbine is fuelled from the utility gas supply and drives a generator. Heat recovery is effected from the exhaust of the turbine and utilised for space heating or process use. The exhaust temperature from a gas turbine can be increased by combusting fuel directly in the exhaust manifold (as with some aviation gas turbines). Birmingham University [8] provides a good example of a gas turbine CHP installation.

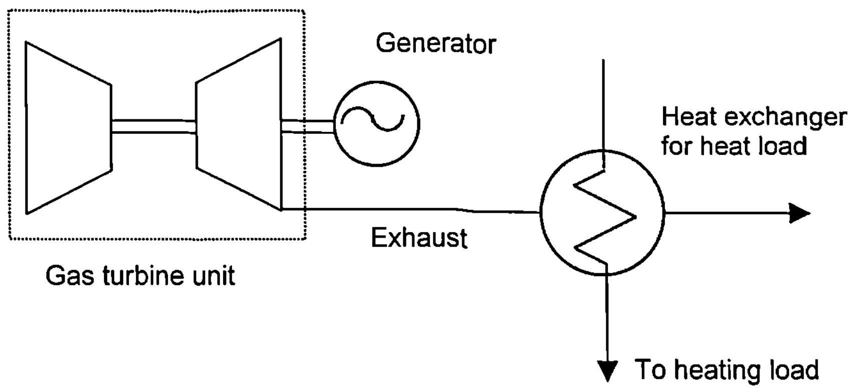


Figure 2.5 Gas Turbine CHP Plant

The waste heat from a gas turbine is of sufficient quality to be able to raise steam at a high enough pressure to drive a steam turbine. This technique (CCGT) of combining a 'topping' gas turbine cycle and a 'bottoming' steam turbine cycle, has a far higher conversion efficiency than would be possible with any single cycle[11,12]. Most CCGT applications are not co-generative, although it is possible to utilise the waste heat from the bottoming cycle for low grade industrial heating purposes. Such an application can be found in the case study of the Kelenföld plant (see Appendix A).

2.3.1.3 Nuclear Reactor CHP Plants

Waste heat from nuclear reactors is used in a number of applications for low grade space heating[12]. Although not of direct interest to this thesis, it serves to contrast different approaches to CHP and the changing nature of the industry. During the 1970s, nuclear CHP was viewed by many European governments as an alternative to fossil fuelled district heating schemes. In particular, former Soviet Block countries implemented some form of nuclear CHP. An example of this is the use of waste heat from one KVLM unit at Pak (Hungary) to heat a nearby town occupied by plant workers. (source: Pak Nuclear Plant Ltd, Hungary) Using waste heat from nuclear reactors is economically viable, but it requires reactors to be built near population centres, and for this reason it could not be applied in the UK. Additionally, development of special purpose nuclear co-generation plants is ongoing [13].

2.3.2 Small Scale CHP Plant - Large Buildings and Small District Heating Schemes

Steam and gas turbine plants are too large to meet the needs of small scale CHP schemes (below 500kWe) economically: for these applications internal combustion engines must be used. The following section reviews the use of internal combustion engines in a CHP mode and examines the necessary alterations and refinements that must be made.

Small scale CHP is readily applied to small district heating schemes and is very common in municipal buildings, such as hospitals and swimming pools.

2.3.2.1 Packaged CHP Concepts

Large scale steam turbine CHP plants are costly to build and run. The space and maintenance requirements for steam plant are high, and given the complexity of large installations, full time staff are employed to control the plant. It would be impractical and uneconomic to install plant of this nature in smaller applications - the *packaged* CHP plant addresses these problems. For a small scale CHP installation to be successful, the following criteria must be satisfied:

- Noise/vibration must be negligible within the host building (or site).
- The plant must be physically unobtrusive, as space may be at a premium.
- The plant must be autonomous in its operation.
- Plant outputs must be compatible with both existing heat and power supplies, as most installations are retro-fitted.

A packaged CHP plant is designed so that all the major components and auxiliary equipment are contained within an acoustically insulated enclosure, fixed by damped mountings. Packaged CHP plants are designed to be stand alone autonomous units, as would be required from a small heating boiler. The case study contained in Chapter 3 analyses the operation of a packaged CHP plant in greater detail. Some characteristics of natural gas fuelled packaged units are shown in table 2.2

Table 2.2 Comparison of Commercial Packaged CHP Plants

(Source: CPS Ltd, www.cpsl.co.uk/products.html : 15/11/98)

Electrical Rating	Thermal Rating	Electrical Efficiency	Combined Efficiency	Engine Type
kW	*kW*	%	%	
110	181	28	78	MAN 6M+g
220	385	30	82	Perkins 12Pg
598	916	30	76	Dorman 12DTg
1050	2100	22	67	Turbomeca GTM

2.3.2.2 Small Scale CHP Engines

All engines employed in small scale CHP are modified conventional stationary generating units or high quality transport units. The CHP plant manufacturer will purchase engines from major manufacturers and make the necessary modifications. Although some small gas turbine units are found in packaged CHP plants (above 400kWe), the majority of manufacturers employ internal combustion engines.

As CHP plant must be durable and reliable, it is necessary that high quality engines are employed. An engine life of over 20 000 hours is desirable for a commercially viable CHP plant (see Section 2.3.2.2.6) and this limits the choice of engine. It is usual for high quality diesel engines to be used (see Table 2.2 for a list of engines used by CHP manufacturers), which may need modifications. The modifications needed depend on the fuel and applications, and are discussed in the following sections.

2.3.2.2.1 Fuel Types

Fuels used by small packaged CHP plants are:

- Diesel
- Natural Gas
- Landfill gas

Those plants that run on gaseous fuel will require extensive cylinder head and fuel system modifications. Spark ignition equipment must be fitted, as gaseous fuel will not undergo a compression ignition cycle. The diesel injector and pump must be replaced by a gas carburettor and intake manifold. The use of gaseous fuel also affects adversely exhaust valves and valve seats, due to high temperatures. These modifications are effected by cylinder head replacement and in some cases, piston replacement. Pre-combustion chambers may be utilised and valve timing adjusted. The use of landfill gas presents further problems due to the quality of supply and requires further alteration of timing.

2.3.2.2.2 Lubrication

As most CHP plants use gaseous fuels, as stated in section 2.3.2.2.1, lubrication requirements will differ from those of the basic diesel engine design. The contents of land fill derived fuels are very aggressive and have a detrimental effect on the engine components, but correct selection of lubrication will reduce such problems. The use of synthetic oils is an area that may be developed to extend the life of gas fuelled engines [14].

2.3.2.2.3 Generators

Most small scale CHP plants employ 3 phase 415V generators, which feed the local low voltage system. Two generator types are employed:

- *Synchronous* generators, that must be synchronised with the utility grid by varying the engine speed.
- *Asynchronous* generators, which can be used as an induction motor to start the engine, in synchronisation with the grid.

2.3.2.2.4 Heat Recovery Equipment

Waste heat recovery by packaged CHP plants is via engine block cooling and exhaust heat exchangers. Exhaust heat exchangers are normally of a simple shell and tube construction, reducing engine back pressure losses. Condensing heat exchangers are manufactured in stainless steel construction, hence increasing the cost over non-condensing designs.

2.3.2.2.5 Control System

The installation of a computer control system is essential for the autonomous operation of a packaged CHP plant. As well as controlling the plant, 'on-board' computers also monitor plant conditions and will alert the servicing company via a modem link. Communication between the control computer and a building energy management system (BEMS) is also possible and aids energy management tasks.

As the engine load drops, so will mechanical efficiency, and at some point it will be uneconomic to run the plant. The load at which a plant should be shut off is a function of engine efficiency and energy costs, which will vary on a yearly basis due to price changes. In the majority of commercial installations, the shut off point is fixed at 50% of full electrical rating.

2.3.2.2.6 Maintenance

As packaged CHP plants are stand-alone devices, servicing is only required at specific intervals, or when a fault occurs. Maintenance is carried out by the manufacturer or by a subcontracting company. Regular service interval for packaged IC based CHP plant are typically between 600 and 2000 hours of running [15]. Major engine overhauls are required ever 20000 to 50000 hours, depending on engine size [16]. Maintenance costs are usually expressed in terms of cost per unit of electrical production (£/kWhe), but as a 38kWe plant requires almost as much maintenance as a 400kWe machine, maintenance unit costs are higher for smaller plants. Maintanance cost are calculated to be in the region of £0.003/kWh to £0.02/kWh [15,16].

2.3.3 Domestic Scale CHP

The previous section has discussed large and small scale CHP - the following section will cover domestic scale CHP (below 1.5kWe). A domestic CHP plant will be defined as a co-generation plant delivering energy to a single dwelling and located in or near the dwelling. Domestic scale CHP is not commercially available in the UK. This section will review research undertaken in domestic scale CHP, which has tended to concentrate on engine issues.

2.3.3.1 Commercial Plants

Only two CHP plants have been commercially available that could be considered as domestic CHP plants in the broadest sense:

- FIAT TOTEN - A small automotive derived FIAT engine was used as the basis for a 15kWe CHP plant (the TOTEN), and was commercially available in the early 1980's [16]. Due to the automotive quality of the engine, plants were known for their poor reliability. Such plants have been installed in hotels and large homes.

- Ecopower (of Biel-Bienne Switzerland) - offered a 5kWe unit based on a Briggs and Stratton single cylinder spark ignition engine. The plant is packaged and controlled in a similar way to small scale CHP plants.

2.3.3.2 Domestic Stirling Engine Research

Stirling engines have a number of characteristics conducive to a good domestic CHP installation:

- High mechanical efficiency of above 40%[18].
- Long engine life (in the region of 50 000 hours) and high reliability [19].
- Low noise and vibration levels [18,19,21], compared to internal combustion engines.

Research work has included improving cycle thermodynamics and drive mechanisms. Much of the commercial interest has concentrated on the use of Stirling engines as an automotive engine. Lately, a number of research institutions and companies have become interested in Stirling engines for domestic CHP application.

British Gas and *Sustainable Engine Systems* [20] are working on commercial Stirling Engines for domestic CHP plants. However, due to industrial secrecy, progress cannot be assessed.

2.3.3.3 Domestic Fuel Cell Research

Fuel cells would form an excellent basis for a domestic CHP plant: as they are electrochemical devices, there no moving parts and hence no noise or vibration problems. As fuel cells are not heat engines, they are not subject to the Carnot limit, so higher conversion efficiencies can be achieved. Work is underway by a number of companies, including British Gas, Oska Gas and Mitshibisi, to develop fuel cell CHP plants. These plants are designed to meet small industrial, not domestic, energy requirements[22,23].

2.3.3.4 Commercial Feasibility of Domestic CHP

Previous academic research [24] has assessed the commercial feasibility of domestic CHP, by examining the likely cost and operating conditions of a small gas engine plant. Different commercial and political scenarios were examined. The cost of plant was found to be in the region of £1000, equivalent to a high quality domestic gas boiler. The following conclusions were made:

- Poor payback periods due to low plant utilisation: owing to the diverse profile of domestic energy demands, plant utilisation would be very low by commercial standards, giving payback periods of over 8 years. This issue will be re-analysed and addressed in chapter 4.

- Due to poor plant utilisation it would uneconomic for energy companies (such Regional Electricity Companies - RECs) to own domestic CHP and sell the energy to the householder.

- A Carbon Tax would have a minimal effect on domestic scale CHP.

- Environmental costing of carbon dioxide emissions reduces the payback period to 3 years. This highlights the environmental advantages of domestic CHP.

Although this exercise did not find domestic CHP economic, it demonstrated that a high plant utilisation is critical. It also found that if plant utilisation could be raised, then the plant would be cost effective.

It was also argued that commercial economic analysis is not applicable to domestic CHP, as a householder will purchase large domestic items such as a freezer or cooker simply because they are needed. Under this assumption, the barrier to domestic CHP implementation becomes more a psychological problem than a technical one.

These finding were similar to conclusions made by a previous study carried out ETSU [25], except for the rejection of REC ownership scenario.

2.3.3.5 Heat Pump Research

Much work has been carried out on various aspects of heat pumps. However, this thesis will only concentrate on the relevant area of Stirling cycle and gas engine driven heat pumps.

Conventionally, heat pump compressors are driven by electric motors. The type of drive for heat compressors has been the subject of both academic and industrial research [26,27]. Engine driven heat pumps have been viewed as an inexpensive alternative to electrically driven devices, when comparing the cost of fuel and utility supplied power. A number of companies have produced engine driven heat pumps for space heat or cooling applications, including Ford and ABB. Plants have been relatively large (over 40kW), utilising large diesel or gas engines/ turbines.

Some research has been previously carried out to develop domestic scale engine driven heat pumps. In 1983 a small company associated with EATS developed a purpose built engine/ heat pump system aimed at replacing domestic gas boilers [28]. This work was taken over by British Gas and has since been discontinued.

British Gas is known to be currently developing CHP technology aimed at the light commercial and domestic markets. Small gas turbine and Stirling engines are under consideration as prime movers.

2.4 Market Development of CHP

The following sections summarise the commercial development of CHP and examine the effects of political policies on the CHP industry. The history of cogeneration in the UK has been extensively covered in the literature [29]. Post War Developments

The 1947 electricity act [31] required the central electricity board, which was to become the CEGB (Central Electricity Generating Board), to investigate methods of utilising waste heat from power stations for domestic or industrial uses. This doctrine, which was intended to be fundamental to the CEGB policies, was subsequently ignored. For technical, institutional and possibly psychological reasons, discussed in the following section, CHP development in the UK was stunted until the 1980's.

In 1957 the Battersea Power station was completed and in accordance with the appropriate tenet of the 1947 electricity act, the station had a co-generation capacity. Waste heat from the station was used in a large district heating scheme, heating over 2,400 flats. This was feasible, due to its inner city location.

The fact that Battersea power station supplied heat to flats should be noted. Dwellings within tower blocks are generally heated by central heating systems and not by individual plants, as with individual dwellings. District heating schemes are readily installed in apartment blocks due to centralised building services, while the distributed supply of heat to individual suburban dwellings is more costly and complex. British cities and large towns, unlike their continental counterparts, are suburban in their character, with the majority of the population living in small separate dwellings. With the perceived failure of the high rise housing programs of 1960s, the option of widespread CHP implementation became nonviable. In continental Europe and the former Soviet block, the high rise nature of urban housing and the social acceptance of inner city power stations allowed CHP to develop widely.

The status quo described was also compounded by institutional problems that prevented the development of small scale CHP schemes. In 1975, guidance documentation was published (G26) that allowed private generators to be connected to the electricity grid. Compliance with these guidelines and a whole host of time consuming related bureaucracy rendered small commercial CHP schemes unfeasible. Restrictions on the use of natural gas also reduced the scope of CHP implementation via gas engines. The price of a *firm* gas supply was tied to the cost of alternative fuels, such as diesel. The intransigence of local area electricity boards in giving permission for grid connection, combined with the above obstacle effectively gave the CEGB a monopoly in supplying power.

2.4.1 Industrial Restructuring

During the 1980's, changes in the UK energy industry regulation removed many of the institutional obstacles to the implementation of co-generation [30,17]. Three fundamental policy changes allowed for the widespread implementation of CHP to start:

- The 1983 Energy Act obliged the electricity industry to encourage the use of CHP. The bill also required energy tariffs to be published, allowing the advantages of small scale CHP to become financially obvious.
- The easing of restrictions in the use of natural gas in generating systems in May 1989 severed the link between the price of gas (for generating use) and alternative fuels.
- The 1989 Electricity Act had many far reaching effects on the electricity industry as a whole, but it had a major influence on the subsequent development of small scale CHP. The introduction of commercial competition in the electricity industry made it easier for commercial concerns to invest in CHP.

2.4.2 Market Penetration of CHP

Over the last 20 years the successive deregulation of the energy industries and advances in computer control systems has led to the widespread implementation of industrial and non-industrial CHP. Small packaged CHP plants have been installed in many hospitals, swimming pools etc., while large CCGT based plants have been built for industrial use. The trend in the market within the UK is for small packaged units supplying a single building, as the problems of supplying heat to large suburban areas still remains.

2.4.3 Environmental Policy

Recent concern over global warming associated with carbon dioxide emissions has led to international action to reduce these emissions. The UK government recognised the potential that CHP has in reducing emissions and is seeking to further increase the installed capacity of CHP to 12 GW, by 2010 (source: CHP-A). The environmental aspects of CHP will be covered in more detail in the following chapters.

2.5 Summary of Review

- Large scale CHP is primarily used in steam turbine plants and is widely used in continental Europe.
- Small scale gas engine packaged CHP plants are readily accommodated and operated within large buildings.
- Domestic scale CHP for the UK is not currently commercially available, with research work concentrating on the use of Stirling Engines.
- Domestic gas engine driven heat pumps have been the subject of commercial and academic research, although no unit is commercially available.
- The development of CHP in the UK has been hampered by institutional problems and the architectural character of British cities.
- Recent political changes have allowed the widespread implementation of small scale packaged CHP plants.

3 Experience with Queens Building CHP

3.1 Introduction

The following chapter is a case study that examines operational, economic and environmental aspects of the Queens Building CHP plant (see Figure 3.1). The study was carried out in order to gain practical experience with small scale CHP, prior to research into domestic CHP.

The Queens Building at DeMontfort University provides startling and innovative accommodation for the School of Engineering. Several energy-saving features are fundamental to the design, including passive ventilation, enhanced day lighting, lighting control and in-house CHP. The CHP plant is an investment for research, teaching and environmental purposes, as well as for long-term financial benefit. The CHP plant installed in the Queens Building was manufactured by Combined Power Services Ltd (Manchester), using a Ford 4FG engine (see figures 3.1 and 3.3). It is located within an internal plant room, alongside a conventional boiler plant. The CHP unit is designed as the priority source of heat in the building. Results and analysis refer only to the CHP running periods [32].

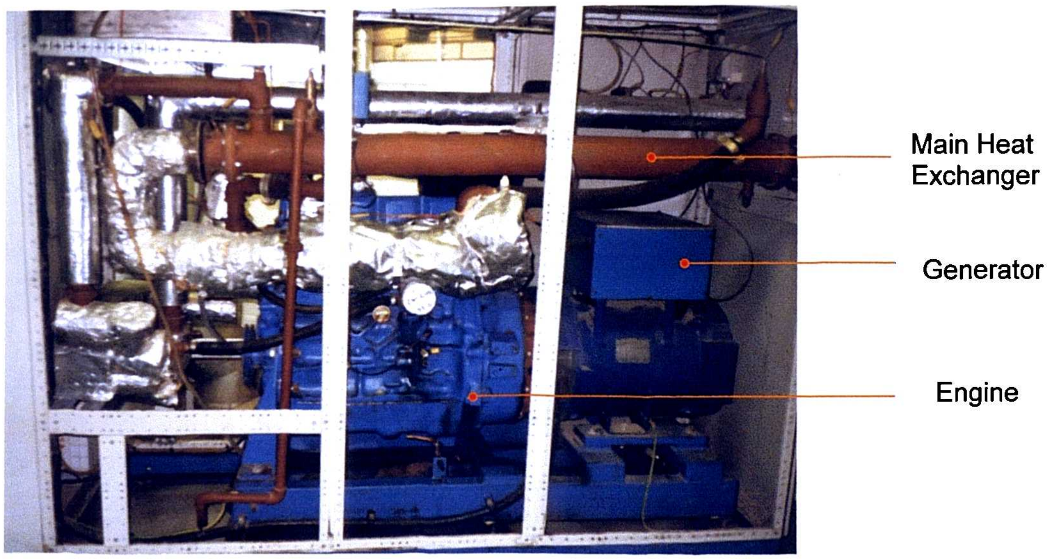


Figure 3.1 The Queens Building CHP Plant

3.2 Plant Installation and Control

The CHP plant, along with the other heating plant in the building, is controlled by the BEMS. The CHP plant acts as the lead boiler of the low pressure hot water (LPHW) heating system for the building, with a condensing boiler and two conventional boilers (see Figure 3.2).

When there are concurrent thermal and electrical demands, the CHP plant will be allowed to operate. The boilers are used to supply additional heat, should the thermal requirement be larger than the thermal rating of the CHP plant. Additional electrical demand is satisfied by the utility supply. Should the electrical or thermal demand drop below the ratings of the CHP plant, then the plant is throttled to reduce its output. If electrical demand is less than 50% of the CHP plant's rating, the plant is not operated (see Section 2.3.2.2.5).

Figure 3.3 illustrates the layout of the CHP plant. The engine is fuelled by utility supplied natural gas (see Section 2.3.2.2.1). The LPHW system is initially passed through a primary heat exchanger where it cools the engine coolant. Exhaust gas from the engine passes into a mild steel shell and tube heat exchanger, where it gives up heat to the LPHW system. The reduced exhaust gas temperature is above 100°C to avoid condensation and hence corrosion (see Section 2.3.2.2.4). The engine drives a synchronous 3 phase generator, at 415V. The package is fitted on a suspended raft to attenuate vibration.

The installation of the CHP plant exhaust ducting caused a number of maintenance problems. Large amounts of condensate and oil residue built up within the exhaust ducting, due to duct layout and inadequate condensate drainage. The cumulative build up of condensate and oil residue resulted in a high exhaust back pressure, to the detriment of plant performance.

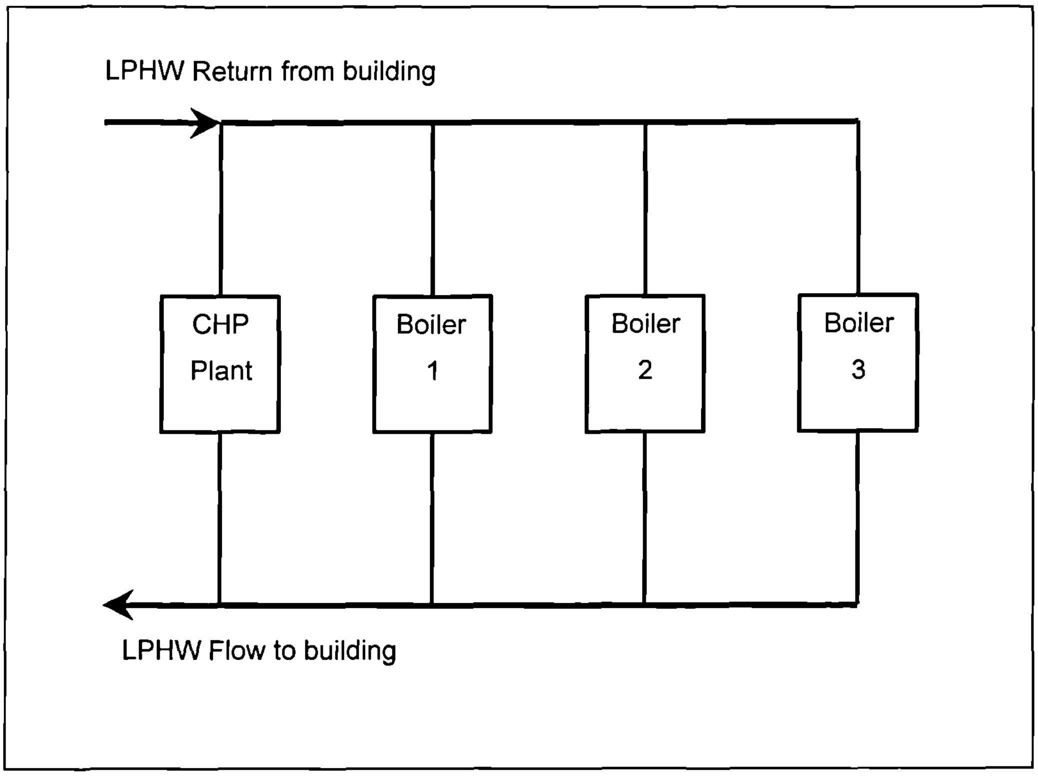


Figure 3.2 Queens Building Heating Plant

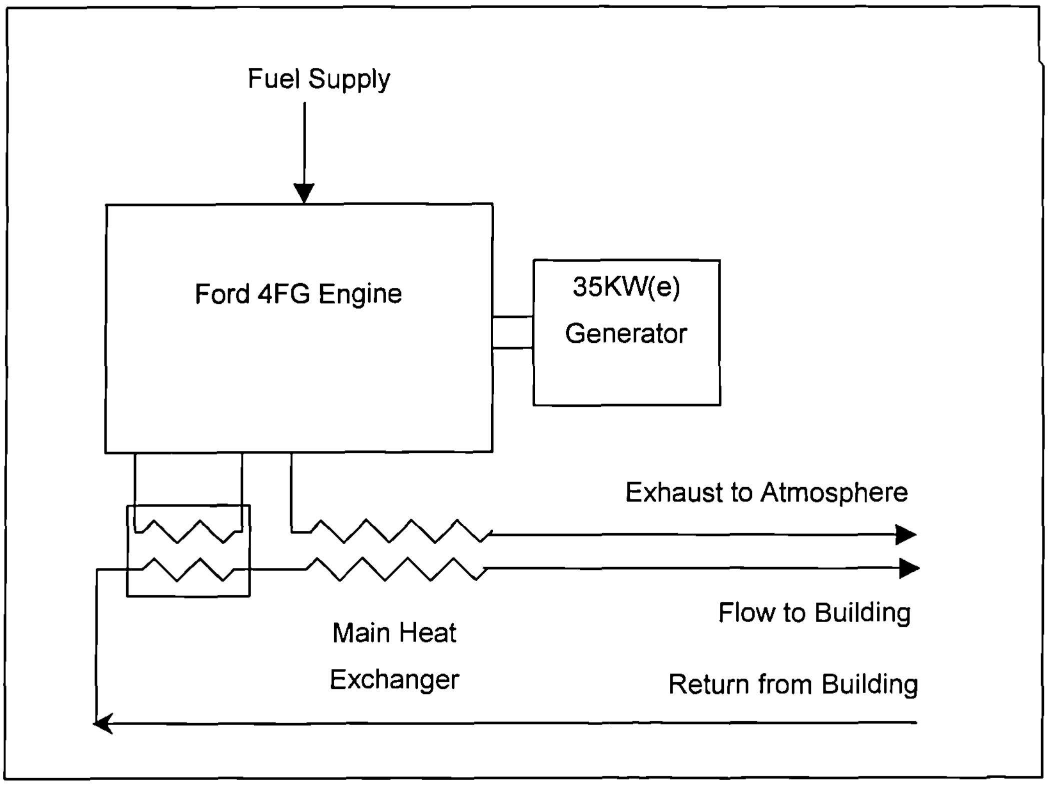


Figure 3.3 Layout of the Queens Building CHP Plant

3.3 CHP Plant Analysis

In the following section, a comparative examination of methods of plant analysis is made. The thermodynamic theory will be developed and then BEMS measured data will be used to ascertain actual plant performance and to compare with the manufacturer's ratings.

3.3.1 Analytical Methods

Traditionally, the analysis of CHP plants has only for allowed immediate outputs of electricity and thermal energy delivered into the heating system [33]. A more exact approach, for CHP plants situated within a building, allows for the complete passage of the energy through the building to the environment (an *in-house* approach). This approach includes the benefit of casual thermal gains. Figure 3.4 illustrates these energy flows from the CHP plant through the whole building.

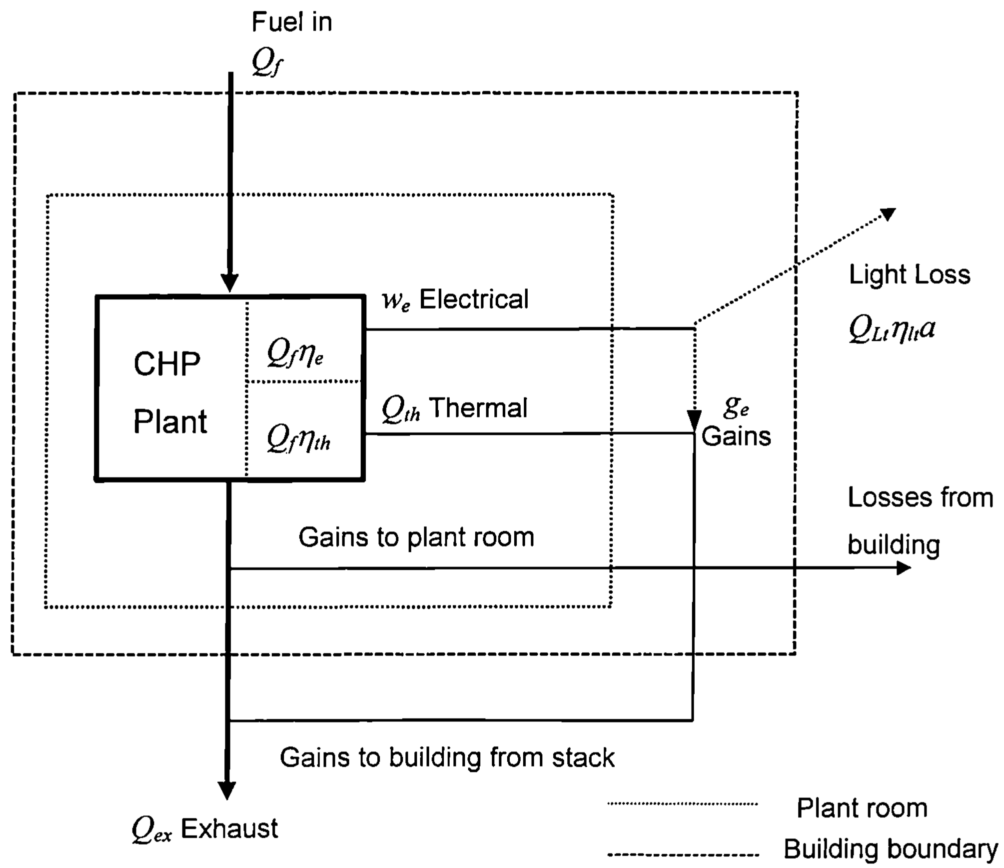


Figure 3.4 Energy Flow From CHP Plant Through Building

The thermal efficiency (η_{th}) is defined as

$$\eta_{th} = \frac{Q_{th}}{Q_f} \qquad (3.1)$$

Traditionally, CHP plant thermal output (Q_{th}) has been considered to be the amount of thermal energy transferred from the plant to the heating water, i.e.

$$Q_{th} = \dot{m}cp_{wt}(T_{out} - T_{in}) \qquad (3.2)$$

Where T_{out} and T_{in} are the flow and return temperatures of the LPHW system, with respect to the CHP plant. Hence,

$$\eta_{th} = \frac{\dot{m}cp_{wt}(T_{out} - T_{in})}{Q_f} \qquad (3.3)$$

The conventional electrical efficiency (η_e) considers the amount of electrical power (w_e) delivered to the electrical system of the building, viz.,

$$\eta_e = \frac{w_e}{Q_f} \qquad (3.4)$$

Hence, the total CHP plant efficiency is conventionally defined as

$$\eta_{chp} = \frac{w_e + Q_{th}}{Q_f} \qquad (3.5)$$

The electricity generated by the CHP plant and utilised in the building will degrade to heat (free or casual gains), apart from light lost through windows as radiation. The light loss will depend on the electrical demand of the lighting system (Q_{Lt}), the luminous efficiency of the lighting (η_{Lt}) and the proportion of light transmitted through windows (α). Hence, the thermal gain from the electricity is

$$g_e = w - (Q_{Lt}\eta_{Lt}\alpha) \tag{3.6}$$

The internal heat gain from the CHP plant room (g_{pr}) into the building, via conduction through the plant room wall, is

$$g_{pr} = U_{pr}a_{pr}(T_{pr} - T_{building}) \tag{3.7}$$

The thermal efficiency now becomes

$$\eta'_{th} = \frac{\dot{m}cp_{wt}(T_2 - T_1) + g_e + g_{pr}}{Q_f} \tag{3.8}$$

or

$$\eta'_{th} = \frac{Q'_{th}}{Q_f} \tag{3.9}$$

where:

$$Q'_{th} = \dot{m}cp_{wt}(T_2 - T_1) + g_e + g_{pr} \tag{3.10}$$

The total plant efficiency, according to the alternative analysis, is therefore

$$\eta'_{total} = \frac{w_e + Q'_{th}}{Q_f} \tag{3.11}$$

3.3.2 Comparison of Analytical Methods

The theoretical analysis was applied to BEMS data to assess the plant performance at full load, which is its normal operating condition. Using the data provided by an energy survey[34] and site plans, the amount of energy lost through windows as light was estimated at 1kW with a lighting efficiency (η_{Lt}) of 20%. The otherwise unheated plant room air temperature was on average 1°C higher than that of the building. Knowing the U value of the building materials and the room dimensions, the fabric gains in this building were found to be 20W, which is negligible. Likewise, in this case, the casual gains from the stack were negligible. Table 3.1 shows a comparison of the results of both the conventional and in-house analysis methods.

Table 3.1 Comparison of Analytical Methods

Analysis	η_{th}	η_e	η_{tot}
Conventional(%)	50.7	27.5	78.2
'in-house'(%)	77.9	27.5	103.6

The total efficiency calculated using the in-house method (η_{tot}') is the apparent efficiency within the building boundary when casual gains are taken into account. The main casual gain of 39kW arises from dissipation of the electricity as useful heat (predominately computers and lights), as shown in Figure 3.4. The conventional analysis showed that the CHP plant is performing according to the manufacturer's ratings. Table 3.2 compares the manufacturer's specified ratings to the CHP plant's insitu performance.

Table 3.2 Comparison of Manufacturer's Stated and Actual Performance

	Q_f	Q_{th}	w_e	η_{th}	η_e	η_{chp}
PERFORMANCE	KW	kW	kW	%	%	%
RATED	138	70	38	50.7	27.5	78.2
ACTUAL	140	68.2	40	48.7	29	77.2

3.4 Plant Utilisation

The load factor (L) of the plant is the (measured) actual electrical energy produced over an extended period, divided by the potential electrical energy produced from continuous operation over that same period, i.e.:

$$L = \frac{q_e}{8760.w_{\max}} \qquad (3.12)$$

In this case for one year,

$$L = \frac{q_e[kWh / year]}{38[kW].8760[h / year]} \qquad (3.13)$$

Currently the CHP plant is providing 12% of the electricity demand within the Queens Building and some 10% of the heating demand (with an annual load factor of 30%). The CHP plant is well utilised during the heating season, meeting heating demands and running at nearly full load from 9am to 9pm, giving a load factor of nearly 50%. During the summer, the load factor is reduced to 15% as only the domestic hot water demand is present. This yields an average annual load factor of 30%, which is low by conventional standards. The University is investigating how to increase summer base load by the supply of heat to other buildings nearby, which would significantly increase the load factor.

3.5 Economic Analysis

In the following section, the economic criteria used to assess CHP plant performance will be defined and evaluated from BEMS measurements. The PBP, and NPV are defined, respectively, as

$$PBP = \frac{C}{savings} \qquad (3.14)$$

$$NPV = -C + \frac{\sum_0^n savings}{(1+r)^n} \qquad (3.15)$$

The useful thermal energy generated (q_{th}) in time t relates to the fuel input rate (Q_f) and the thermal efficiency (η_{th}) of the unit, i.e.

$$q_{th} = \eta_{th} Q_f Lt \qquad (3.16)$$

The value of this thermal energy is that of the equivalent cost (C_{boiler}) of using a boiler to produce the same amount of useful heat.

$$C_{boiler} = \frac{q_{th} c_f}{\eta_{boiler}} \qquad (3.17)$$

The annual fuel cost (C_f) to generate q_{th} from the CHP plant is

$$C_f = \frac{q_{th} c_f}{\eta_{th}} \qquad (3.18)$$

The increased cost of fuel is (C_f - $C_{bloiler}$).

The value of the electricity generated by the CHP plant (C_e) is at the same unit price of the grid electricity supply,

$$C_e = q_e c_e L \tag{3.19}$$

The maintenance cost of the CHP plant is

$$C_m = q_e c_m L \tag{3.20}$$

So the net saving becomes

$$savings = (C_{boiler} + C_e) - (C_m + C_f) \tag{3.21}$$

where (C_{boiler} + C_e) are the energy costs without CHP.

This procedure was applied to measured BEMS and other data to assess the economic performance over one year. The capital cost of the machine does not include maintenance or additional instrumentation for teaching and research. Maintenance cost (c_m) is a standard amount from other experience[16]. Table 3.3 presents the discounted cash flows.

The following values are used in the analysis:

capital cost (C) = £28 500,
gas unit cost (c_f) = £0.010/kWh,
electricity unit cost (c_e) = £0.065/kWh,
maintenance unit cost (c_m) = £0.0125/kWhe,
annual load factor (L) = 0.296,
time (t), 1 year = 8760h,
discount rate (r) = 5%.

Substituting in equations

q_{th} = 8760 x 138 x 0.296 x 0.49 = 174262 kWh/y,

C_{th} =174262 x 0.010/0.8= £2178 /y,

C_f=8760 x 138 x 0.296 x 0.01= £3578/y,

C_e = 8760 x 138 x 0.296 x 0.29 0.065 = £6745/y,

C_m = 8760 x 138 x 0.296 x 0.29 x 0.0125 = £1324/y,

savings = (2178 + 6745) - (1324 + 357) = £4021/y.

PBP = 7.1 y

Table 3.3 Discounted Cash Flow

Year	Discounted Cash Flow	Sum	NPV
	£	£	£
0	4021	4021	-23979
1	3830	7851	-20149
2	3647	11489	-16502
3	3473	14971	-13029
4	3308	18279	-9721
5	3151	21430	-6570
6	3001	24430	-3570
7	2858	27288	-712
8	2722	30010	2010

3.6 Environmental Performance

In the following section, the environmental performance of the CHP plant will be assessed. The environmental benefits of the CHP plant will be quantified in terms of reduced carbon dioxide emissions and associated removal costs. The abatement costs will be used to recalculate the payback period of the CHP plant.

3.6.1 Environmental Analytical Method

The following analysis will examine the emissions evolved as a consequence of the Queens Building's energy consumption under different plant scenarios. The emissions that would be evolved with conventional forms of energy delivery are used as a reference. Carbon dioxide emissions are incurred both directly, through the use of the gas fired boiler plant, and indirectly through electricity consumption (as a consequence of fossil fuel combustion in utility power plants). The concurrent generation of both heat and electricity via the CHP plant leads to reduced fossil fuel consumption and hence lower carbon dioxide emissions. CHP generated energy *displaces* utility derived energy and the associated emissions, hence the effect is a reduction in carbon dioxide emissions. The CHP plant may not be able to meet all the building's energy demand and some conventional energy supplies must be utilised to meet the remaining (adjusted) demand, which will have associated emissions. Figure 3.5. illustrates the method (see appendix C for a detailed derivation of carbon dioxide emissions calculations).

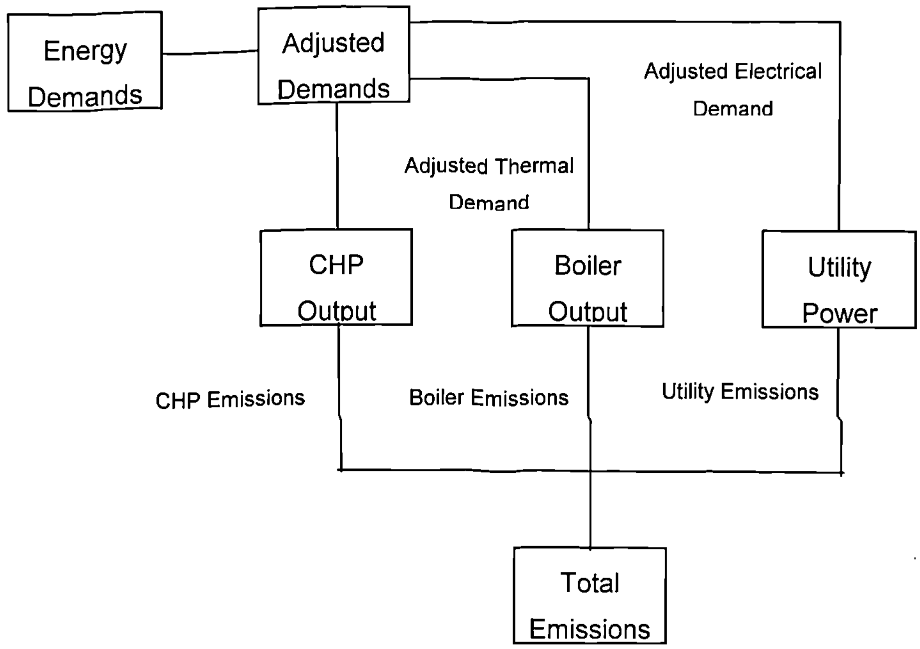


Figure 3.5 Carbon Dioxide Emissions Method

3.6.2 Environmental Results

The CHP generated electricity displaces energy delivered from the grid and from conventional boiler plant, and hence displaces emissions evolved from such sources (see Figures 3.6 and 3.7). Table 3.4 summarises the environmental effects, comparing the emissions due to energy consumption from conventional sources with that from CHP energy production, over one year at current demand rates. The net effect is a reduction of the emissions of most gaseous species when CHP is used.

Carbon dioxide emissions are significantly reduced due to:

- *High CHP plant efficiency* - the relatively high efficiency of the CHP plant reduces fuel consumption. The CHP plant may be viewed as an boiler that produces electricity without any added emissions.
- *Natural Gas fuelling* - Natural gas contains less carbon per unit energy than coal, which is the primary fuel used in utility power generation, hence the carbon intense coal produces more carbon dioxide per unit power than natural gas.

The carbon dioxide emissions depend heavily on the mix of plant and fuel used by the utility power stations. Sulphur dioxide (SO_2) and hydrocarbon (C_xH_x) emissions are virtually eliminated with natural gas fuelled CHP, as the fuel contains practically no sulphur and non-methane volatile hydrocarbons.

Nitrogen oxide emissions are increased due to the combustion pressures and temperatures associated with an internal combustion engine, while the lower temperatures and pressures required by coal fired steam turbine plant lead to lower NO_x production. The use of a catalytic converter on the CHP plant exhaust would eliminate NO_x emissions.

Table 3.4 Environmental Results

Species	CO_2	NO_x	SO_2	C_XH_X
	kg/y	*kg/y*	*kg/y*	*kg/y*
Non CHP emissions	133 000	480	840	690
CHP emissions	65 000	600	0	2
Reduction	68 000	-120	840	688

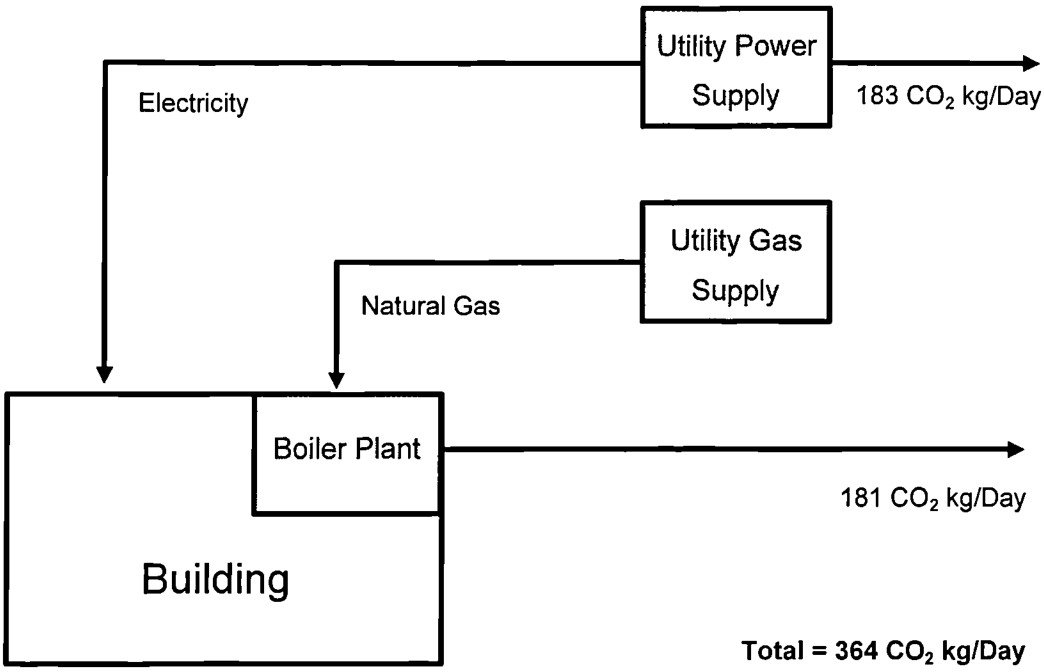


Figure 3.6 Carbon Dioxide Emissions from Conventional Sources

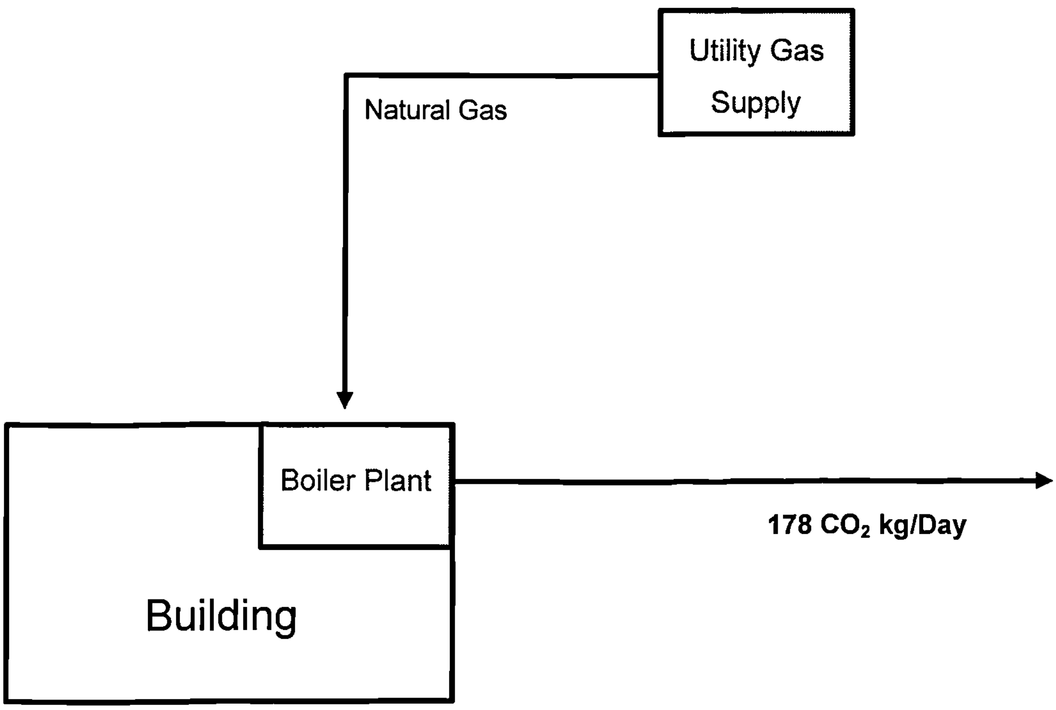


Figure 3.7 Carbon Dioxide Emissions From CHP Plants

3.6.3 Abatement Costs

This section will quantify the environmental benefits of the Queens Building CHP plant in economic terms.

The approach taken is to examine methods of CO_2 removal from the exhaust gas of the generator plant. Owing to economies of scale and plant size, it would currently only be feasible to fit CO_2 "scrubbers" to large centralised plant, and hence in this discussion the removal of CO_2 will only be considered for power stations. In the UK, thermal energy is almost all provided by relatively small boiler plant to which the fitting of scrubbers is not feasible. The fitting of scrubbers to power stations will increase electricity unit prices by additional capital costs and decreasing power station conversion efficiencies. Three types of CO_2 removal are under development. Table 3.5 summarises the unit cost of CO_2 removal and the increase of electricity costs that would be realised, for each of the technologies[24], where cost could be established. In addition to the two technologies cited in Table 3.5, cryogenic separation is also possible. The average unit cost will be used in further analysis. This method of costing CO_2 emissions is more accurate than the more subjective method of assessing environmental damage due to global warming (an external cost). The subject of costing CO_2 emissions is a highly specialised field: other methods can be applied, depending on context and political agenda. Abatement costs do not have any relation to the cost of environmental damage, unlike external costs.

Table 3.5 Carbon Dioxide Removal Costs

System	**Removal Cost** *(£/Tonne)*
Chemical Absorption	93.0
Membrane Separation	80.0
Average	86.5

The CHP plant reduces the Queens Building carbon dioxide emission by 68 tonnes/year (see Table 3.4). By applying the average removal cost, the economic value of the reduced emissions is £5882 /year. When this is taken into consideration, the pay back period is reduced from 7 years to 4 years.

This method emphasises the environmental benefits of energy conservation projects that are not financially apparent with conventional analysis. The abatement costing method can be exploited to justify projects conventionally viewed as "uneconomic" to financial institutions and financiers. Organisations using this criteria will be encouraged to invest in environmentally positive projects.

3.7 Inference

One of the objectives of this study of the Queens Building was to gain experience with small scale CHP so that relevant inference could be drawn to aid domestic scale CHP work. The following section will summarise the experience gained.

3.7.1 Load Factor

The load factor of a CHP plant is critical to its economic and environmental effectiveness. The Queens Building CHP plant has a very low annual load factor of approximately 30%, which would be considered uneconomic in commercial terms. As conventional financial analysis indicated, the project is "uneconomic", with a 7 year pay back period calculated from energy cost savings, although the project had a number of non-financial advantages such as research and teaching opportunities. By raising the load factor to 50% the payback period would be reduced to under 4 years. This highlights the criticality of maintaining a high load factor (see Table 3.6). In order to maintain a high load factor, a CHP plant must be flexible in meeting the demands of a building. This becomes more critical with small buildings, as the variations in demand are greater (both in daily and seasonal periods). Correctly sizing a CHP plant in relation to the building is another key criteria in achieving a high load factor.

Table 3.6 Load Factor Economics

Load Factor	Pay Back Period	NPV
	years	*£*
30%	7.1	2010
50%	4.2	4781
% Improvement	41	138

3.7.2 Reduced Carbon Dioxide Emissions

It has been demonstrated that small scale CHP is an effective method for reducing a building's carbon dioxide emissions, with an effective reduction in the case of the Queens Building of 68 tonnes/year. The equivalent cost of carbon dioxide removal would be in the order of £5882/year. The reduction in carbon dioxide emissions is dependant on fuel type and plant utilisation, which again highlights the importance of maintaining a high load factor.

4. Domestic CHP and the CHP/HP Concept

4.1 Introduction

The following chapter assesses the merits of conventional CHP* for domestic application, and introduces the concept of a combined heat and power plant incorporating a heat pump (a CHP/HP plant). Demand based modelling is used to demonstrate the relative merits of domestic CHP and CHP/HP. Analysis of the model results justifies the development of a prototype plant.

4.2 Domestic Application of Conventional CHP

It will be demonstrated in the following section that conventional CHP is not readily applied on a domestic scale. Further analysis is contained in Section 4.5. Fundamental problems arise from the characteristics of heat engines.

With present commercial technology, only small internal combustion engines and Stirling engines (see Section 2.3.3) are suitable candidates for domestic CHP prime movers. Such heat engines have a heat to power ratio of 2:1, while domestic heat to electrical demand ratios can vary dramatically from 1:1 to 10:1. To maximise the potential economic benefits of domestic CHP, it would be necessary to size a plant on the electrically based load requirements of a dwelling. With domestic base electrical loads of less than 0.5kWe, a very small engine would be required. The thermal output from a CHP plant, utilising such a small engine, would be negligible compared to demand. To increase the thermal output, a larger engine is required. In this case the engine would seldom run at full rated load and low part load efficiencies would negate any environmental or economic benefits of a domestic CHP plant. This point is discussed in more detail with analysis in Section 4.5.

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* i.e. a CHP plant with heat recovery and power output from a single heat engine.

Engine life also presents potential problems, as a small engine will have a very short life and a CHP plant will incur higher maintenance costs. Again it is desirable to employ a larger and more durable engine to increase plant life, but with the same consequence of reduced plant utilisation. A clear mismatch exists between required engine size and plant economics. The solution to both these problems is the incorporation of a heat pump into domestic CHP plant - a CHP/HP plant.

4.3 The CHP/HP Concept

A CHP/HP plant would consist of a conventional CHP plant and an electrically driven vapour compression heat pump. The CHP plant would use a heat engine to drive a small generator, with heat recovery from an engine exhaust heat exchanger and engine cooling system. The engine may be fuelled from the natural gas utility supply, and the generator output would be compatible with the utility grid (allowing for electricity import). The electrical output of the generator can be delivered to the host dwelling and utilised by the heat pump. The thermal output from both engine and heat pump would be used in the dwelling's conventional low pressure hot water heating system. The incorporation of the heat pump would afford a high degree of flexibility to the system, which can run in a number of modes, detailed in Sections 4.3.1 to 4.3.4.

4.3.1 CHP Mode

When electrical demand is greater than the generating capacity, all the generated electricity will be used to satisfy demand and the heat pump will not operate. Thermal output is from the engine only. Hence, the plant operates as a conventional CHP plant (see Figure 4.1).

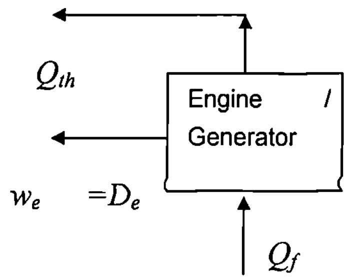

Figure 4.1 CHP Mode

4.3.2 Heat Pump (HP) Heating Mode

With a high thermal demand and no electrical demand, all the generating capacity can be used to drive the heat pump. Heat recovery will be from the engine and from the environment via the heat pump. In this configuration the plant is effectively a gas driven heat pump (see Figure 4.2).

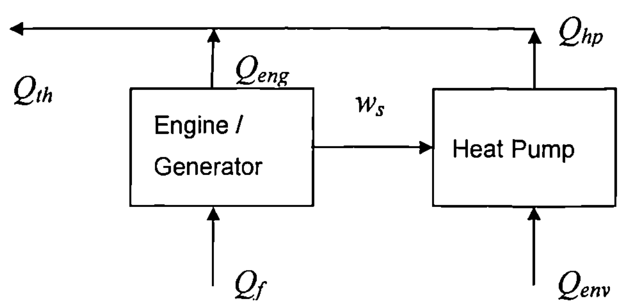

Figure 4.2 HP Mode

4.3.3 CHP/HP Mode

When electrical demand is less than the generator capacity, surplus electricity (w_s) can be used to drive the heat pump, boosting the thermal output if required. This allows the engine to run at a higher load than for conventional CHP. This hybrid CHP/HP operation is illustrated in Figure 4.3.

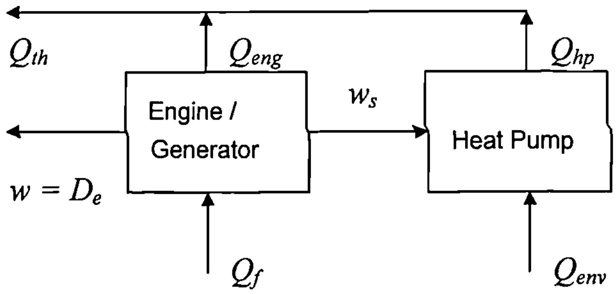

Figure 4.3 CHP/HP Mode

4.3.4 Cooling Mode

If cooling is required, the condenser and evaporator of the heat pump can be reversed, to allow the plant to cool the dwelling as well as providing power, as shown in Figure 4.4.

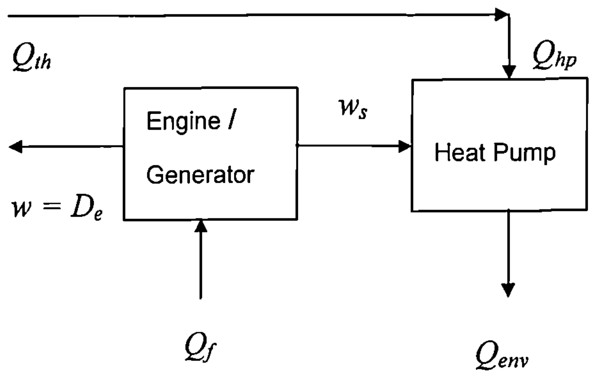

Figure 4.4 Cooling Mode

4.3.5 CHP/HP Plant Utilisation

The load factor (as defined in Section 3.4) of a CHP/HP plant is a function of both electrical and thermal demand, as the thermal output of a heat pump is related to the electrical input. In the CHP/HP mode, the surplus electrical capacity is directed to the heat pump, thus boosting thermal output. To maintain maximum efficiency it would be desirable for the heat pump to utilise all surplus generating capacity. If the thermal demand (D_{th}) is less then the thermal output (Q_{th}), the engine must be modulated to satisfy:

$$D_{th} = Q_{th} \tag{4.1}$$

$$\therefore D_{th} = Q_{hp} + Q_{eng} \tag{4.2}$$

The thermal output of the plant depends on heat recovery from the engine (Q_{eng}) and the heat pump output (Q_{hp}). The heat pump output is dependant on the surplus electrical generating capacity after the electrical demand has been satisfied, hence:

$$D_{th} = w_{\max} L\lambda + (w_{\max} L - D_e)COP \tag{4.3}$$

Rearranging equation *4.3* to find L gives:

$$L = \frac{D_{th} + D_e COP}{w_{\max}(\lambda - COP)} \tag{4.4}$$

As the engine load reduces, part load efficiencies reduce and at some point the plant's operation will cease to be economic compared to conventional forms of energy supply and the plant must be shut down. The load at which the plant shuts down is a function of energy unit prices, plant efficiencies and maintenance cost. For further development of this point, see Section 7.4.1.

4.4 Preliminary Modelling

In order to assess the CHP/HP concept and highlight the inadequacies of conventional domestic CHP application, a simple demand based model was written. Daily domestic electrical and thermal demands on an hourly basis, were used to drive the model, which calculated load factor, plant economics and plant emissions. If the plant operation is uneconomic with respect to conventional forms of energy supply, then the plant is shut down and the utility supply is used. For a detailed explanation of the demand driven model, see Figure 4.5.

4.4.1 Model Assumptions

The demand driven model contains a number of assumptions and simplifications, listed below. These assumptions are addressed in more detailed modelling, described in Chapter 7.

- The model only considers the quantity of energy and not the quality. The temperatures of plant thermal output and environment are not considered, although heat pump and engine thermal delivery will depend on these conditions.

- Heat pump COP is assumed to be fixed at 3:1 and remains constant with respect to heat pump power consumption. The maximum electrical input to the heat pump is assumed to be 1kWe.

- Engine full load efficiency is assumed to be 25% with a maximum rating of 1kWe and a heat to power ratio of 2:1.

- Engine part load efficiency is assumed to be a linear relationship with engine load (see Figure 4.6).

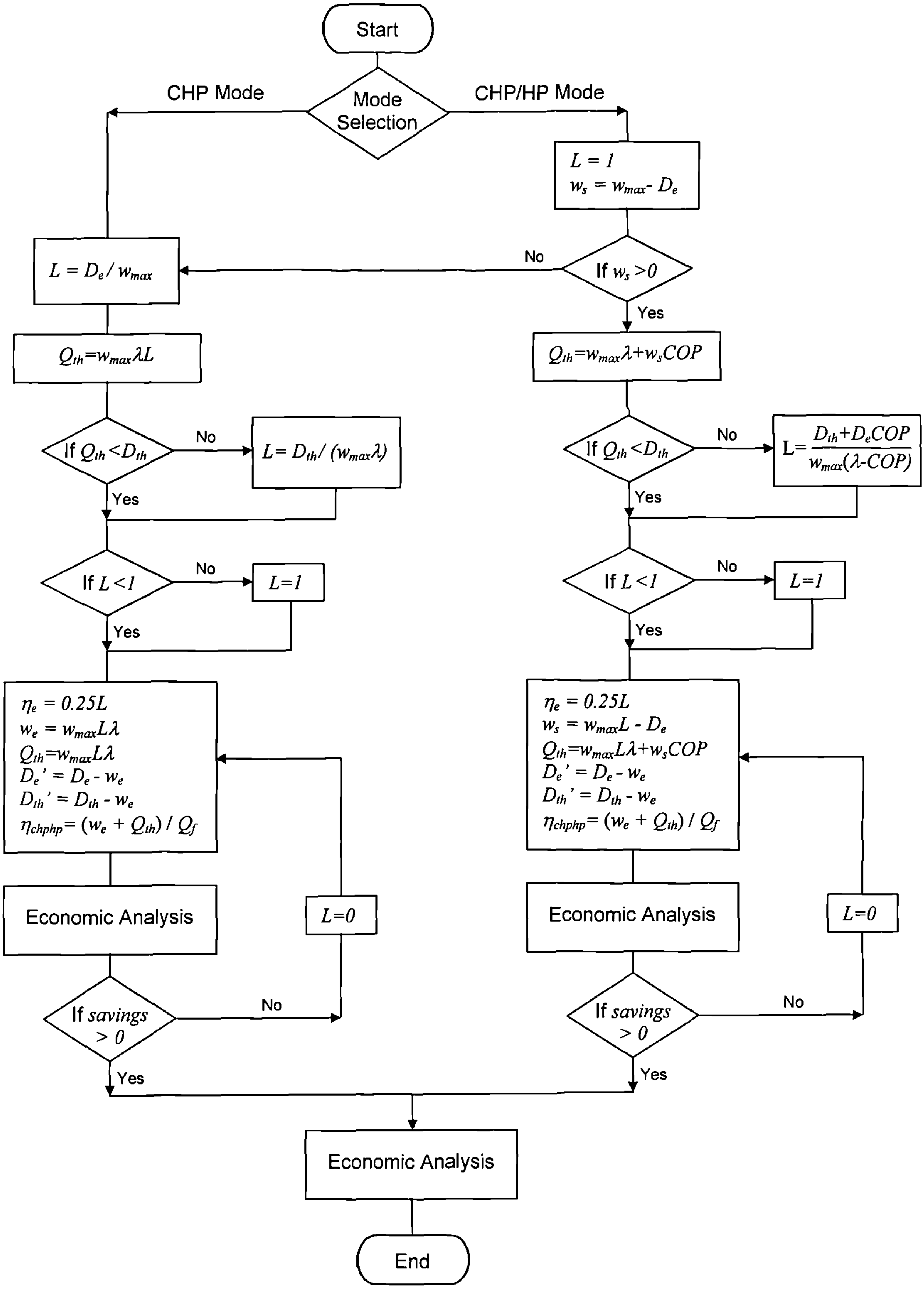

Figure 4.5 Demand Driven Model

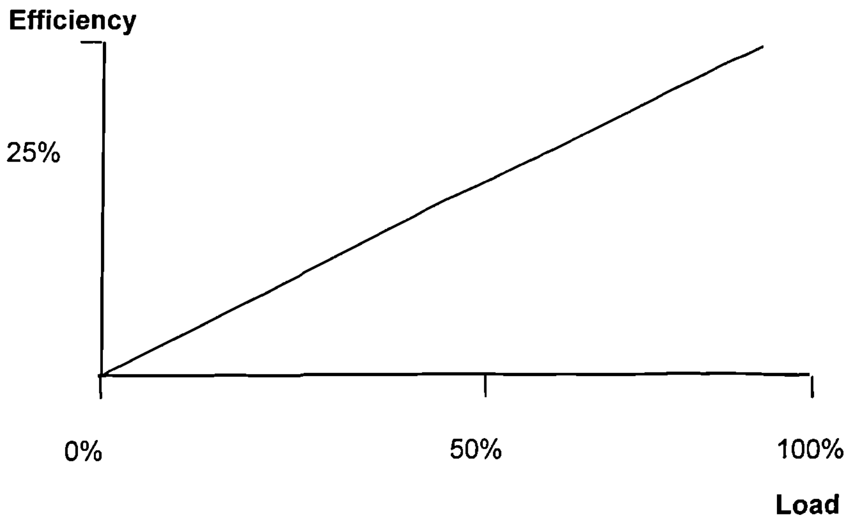

Figure 4.6 Part Load Efficiency of Engine (assumed)

4.5 Analysis of Simulation

Constants were either supplied by component manufacturers or were experimentally derived. Thermal and electrical consumption data for a dwelling over a 24 hour period [39] were used to drive the model. The general results of the simulation are summarised in Table 4.1.

Table 4.1 Results of Simulation

Criteria	CHP	CHP / HP
Total thermal demand met (%)	13.39	75.74
Total electrical demand met (%)	40.29	81.68
Average load factor (%)	19.10	60.42
Average effective efficiency (%)	57.29	86.95
Total financial savings (£/day)	00.16	0.78
Total CO_2 reduction (%)	08.2	33.5

It can be seen that a CHP/HP plant can meet the majority of the energy demands economically and with enhanced environmental benefits. This compares very favourably with a conventional CHP plant of the same maximum electrical rating, which can only meet a fraction of the energy requirement and would have little accumulative environmental benefit. In particular the thermal delivery of a CHP plant is very poor as the small engine required to meet electrical demand efficiently has a relatively low thermal output. This is overcome by incorporating a heat pump.

4.5.1 CHP/HP Analysis

With reference to Figures 4.7 and 4.8, the dynamics of the CHP/HP system can be explained in relation to meeting highly variable thermal and electrical demands. In general, the ability of CHP/HP to meet domestic energy demand is clearly demonstrated. Before 0600 there is not enough energy requirement to run the plant efficiently. At 0700 the thermal demand is very high and all the surplus generating capacity is being utilised by the heat pump (saturated CHP/HP mode). Hence the engine is running at full load (i.e. $L = 1$) where:

$$D_{th} > w_{max}\,\lambda + (w_{max} - D_e)\,COP \qquad (4.5)$$

From 0800 to 1700 the electrical demand is low enough to allow surplus generating capacity to be used by the heat pump. However, lower thermal demand requires that the engine modulates down to an appropriate load factor. In this modulated CHP/HP mode, the engine is under part load (i.e. $L<1$). In this condition the plant efficiently meets both thermal and electrical demands, as:

$$D_{th} = w_{max}\,\lambda + (w_{max} - D_e)\,COP \qquad (4.6)$$

When electrical demand is higher than the electrical rating of the plant ($D_e > w_{max}$), no surplus generating requirement is available for the heat pump and the CHP mode is invoked. The thermal and electrical output for both a CHP and CHP/HP plant are equal for this period.

4.5.2 CHP Analysis

The simulation demonstrates that conventional CHP cannot be applied to a domestic application. The base load of the dwelling is not high enough to allow a CHP plant to run economically. It is only during periods of high electrical demand (see Figure 4.7) that the plant can run economically. To increase the load factor, a smaller engine would have to be employed, which would again raise the problems summarised in Section 4.2.

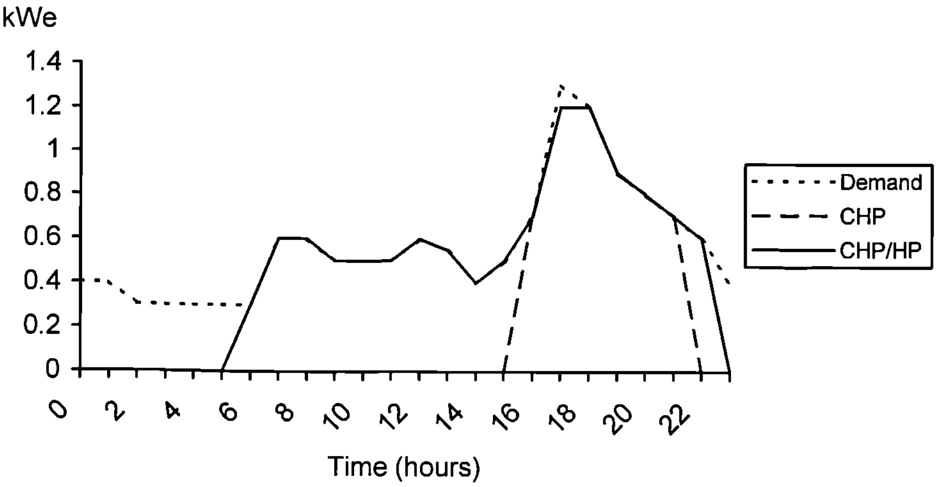

Figure 4.7 Electrical Demand and Demand Driven Model Predicted Supplies

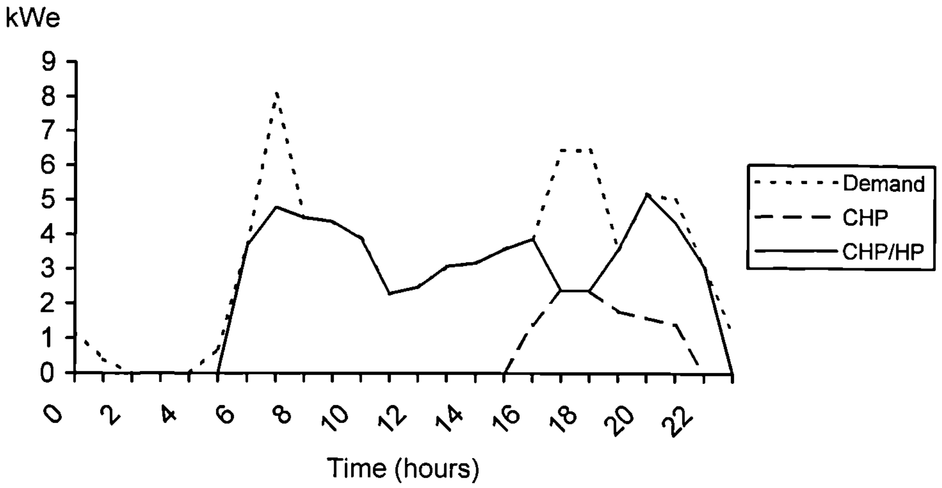

Figure 4.8 Thermal Demand and Demand Driven Model Predicted Supplies

4.6 CHP/HP Plant Optimisation

A CHP/HP plant can be optimised for a number of different criteria, depending on the rating of the heat pump and engine. With the incorporation of a heat pump into the plant, further analysis is required into the relationship between plant components. Demand based modelling is used to find optimum plant configurations for different criteria by varying plant size.

Different demand requirements will have different optimal plant ratings, and as demand requirements vary considerably, a compromise plant configuration must be found that will optimise the plant for a number of demand requirements. The demand profile used in Section 4.5 provides a good example of varying energy requirements and hence will be used in the optimisation analysis. This analysis will not provide definitive plant configurations for domestic CHP/HP, as this will vary according to demand profile, but will serve to demonstrate the differences in plant configuration with respect to optimisation criteria.

The following sections will discuss CHP/HP plant optimisation and analyse the relationship between the engine and heat pump. This analysis is also used to further highlight the advantages of CHP/HP over conventional CHP. Although only environmental and economic optimisations are of practical interest, the optimisation of load factor and energy delivery have direct influence and must be examined first.

4.6.1 Relative Plant Ratings

For maximum benefit, the heat pump electrical rating and engine maximum work output should be the same. If the heat pump were to be larger than the engine, utility supplied electricity would be required, which would render the heat pump operation uneconomic, as:

Unit cost of heat from conventional boiler plant:

$$c_{th} = \frac{c_f}{\eta_{boiler}} \qquad (4.7) \qquad c_{th} = \frac{1.2}{0.7} = 1.71 p / kWh$$

Unit cost for heat pump heating:

$$c_{th} = \frac{c_e}{COP} \qquad (4.8) \qquad c_{th} = \frac{7.2}{3} = 2.4 p / kWh$$

If the electrical rating of the heat pump were to be less than that of the engine, the combined plant would lose flexibility and would not be able to operate as a gas driven heat pump (see Section 4.3.2.) while maintaining maximum engine efficiency. Hence, to maintain flexibility and economic operation, the maximum heat pump electrical rating must match the maximum engine work output.

4.6.2 Load Factor Optimisation

The plant load factor is the primary consideration in the calculation of the other plant criteria. The load factor of a CHP/HP plant is a function of thermal/electrical loads and plant ratings. Figure 4.9 illustrates the relationship between load factor and plant size. Although the demand profile will affect the relationship, the general trend is true for all plant applications.

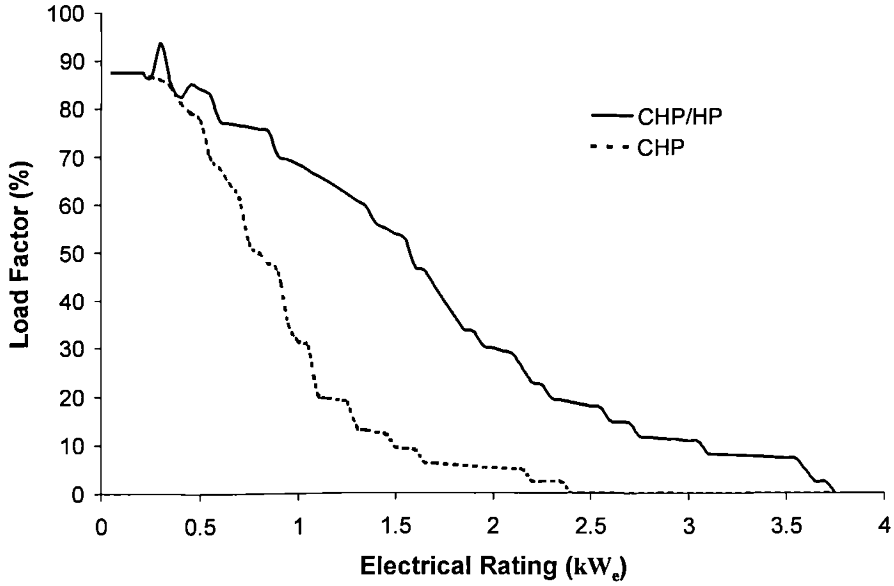

Figure 4.9 CHP/HP Load Factor Optimisation

At a very low plant rating, the plant utilisation is high as the demand will always be greater than the plant delivery. Load factor will remain constant until the plant's energy delivery can exceed the demand. At this point, the load factor starts to fall. Eventually the plant rating becomes too high to maintain economic operation.

The incorporation of the heat pump into the CHP/HP plant allows a plant of relatively high rating to maintain a high load factor compared to a conventional CHP plant (as illustrated in Figure 4.9). Utilising surplus generating capacity in the heat pump allows a larger engine to be used and hence addresses the issues raised in section 4.2. A larger plant will also be able to satisfy a wider range of energy demand requirements. This maximising of environmental and economic benefits will be discussed in the following sections.

4.6.3 Maximisation Of Electrical Delivery

The relationship between plant size and electrical demand satisfied is illustrated in Figure 4.10, for both CHP and CHP/HP plants. As with load factor (see Section 4.6.2), the detailed form of the relationship is due to the varying demand profile. However, the general relationship is correct for all varying demand profiles.

Initially, the percentage of daily electrical demand satisfied is small, owing to the small plant rating. As plant rating is increased, more demand can be met. As plant rating increases further, the load factor decreases (as explained in Section 4.6.2), which leads to reduced output. The conflicting factors of plant size and load factor produce an optimal plant rating.

For low plant ratings, the CHP and CHP/HP relationships are identical, since with low plant ratings no surplus generating capacity is available to drive a heat pump and the CHP/HP plant runs in CHP mode. The coarseness of the relationships illustrated in Figure 4.10 is due to the varying demand profile. Maximum electrical delivery for a CHP/HP plant occurs at a plant rating of 0.85kWe.

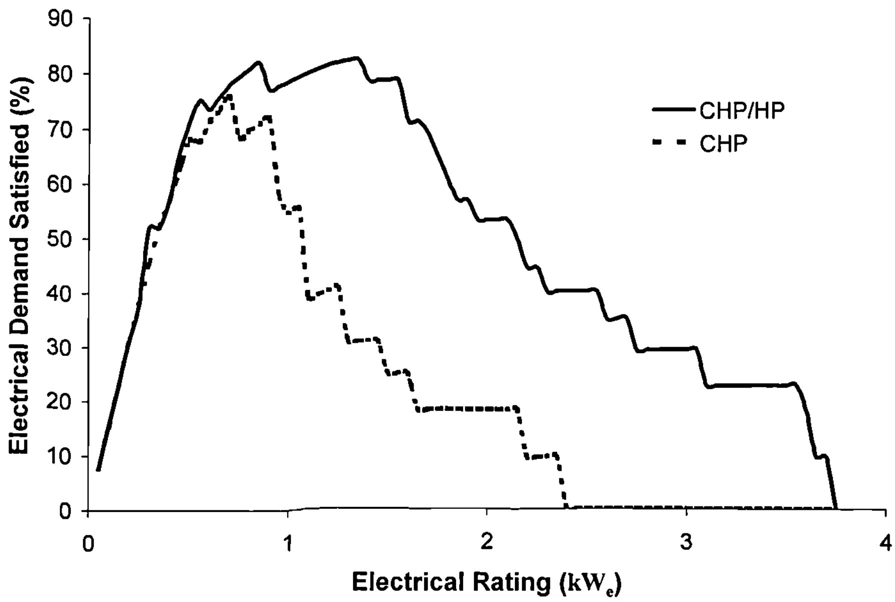

Figure 4.10 CHP/HP Electrical Demand Optimisation

4.6.4 Maximisation of Thermal Delivery

The relationship between thermal delivery and plant rating (see Figure 4.11) is dictated by the same mechanism described in Section 4.6.3, with the conflicting factors of plant size and load factor creating an optimal point. A comparison between CHP and CHP/HP relationships highlights the benefits of heat pump incorporation, with an enhanced thermal delivery.

The optimal CHP/HP plant size for a maximised thermal delivery is larger than that for maximised electrical delivery. A larger plant rating will allow for greater thermal delivery, as more surplus generating capacity is available for the heat pump, enabling the plant to satisfy peak thermal requirements and raising the overall daily thermal output. Increasing plant size further reduces daily thermal output as a consequence of a reduction in load factor (as per the mechanism identified in Section 4.6.3).

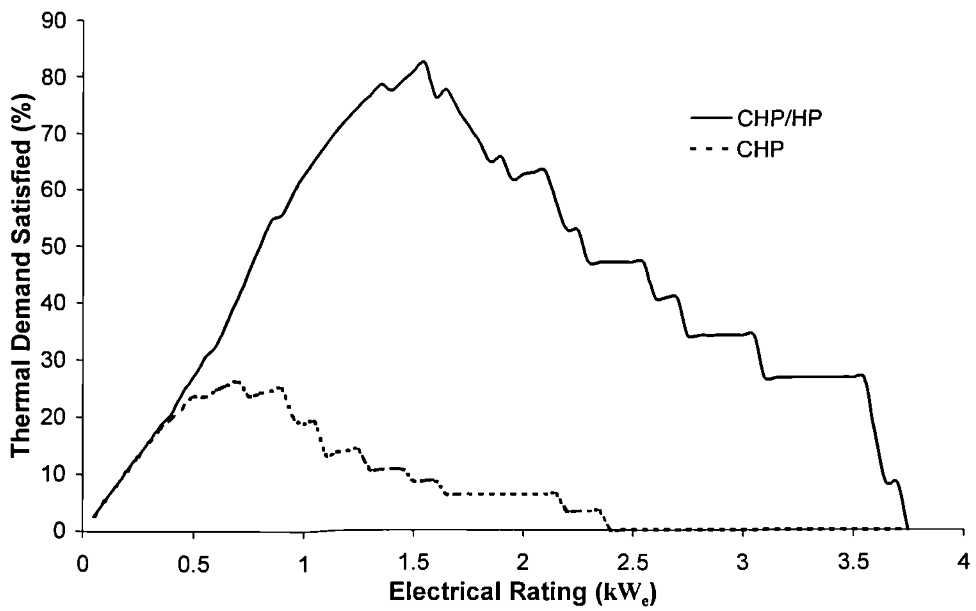

Figure 4.11 CHP/HP Thermal Optimisation

4.6.5 Environmental and Economic Optimisation

Environmental and economic optimal plant ratings are consequential to the criteria discussed previously. Both *environmental and economic performance* have a similar relationship with plant rating, as both are dependant on electrical and thermal delivery, the mechanisms described in Sections 4.6.3 and 4.6.4.

To optimise a CHP/HP plant for maximum environmental benefit, it is necessary to minimise carbon dioxide emissions. The relationship between carbon dioxide emissions and plant electrical rating (shown Figure 4.12) is dependent on the characteristics of the utility energy supplies and energy demands. The displaced emissions (see Section 3.6) caused as a consequence of the usage of utility supplied electricity will vary according to which generating plants and fuels are utilised. The optimum environmental plant rating under the assumed conditions (see Appendix C) is 1kWe.

The optimal plant size for maximised economic benefits is a function of plant efficiencies and utility energy prices: in this case the optimal plant rating is 1.1kWe. With increased utility energy prices, the larger CHP/HP plants would become economic, conversely lower utility energy prices would favour smaller machines.

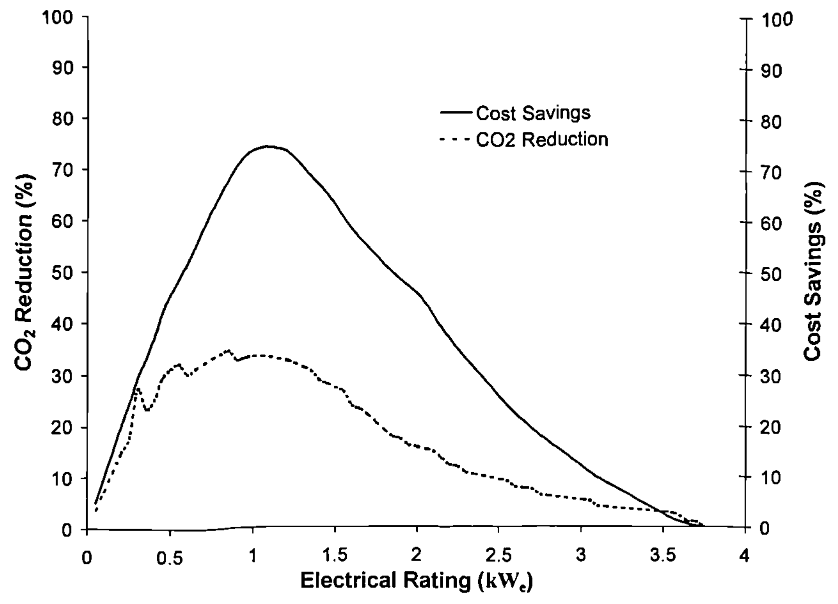

Figure 4.12 CHP/HP Environmental and Economic Optimisation

4.7 Summary

Conventional CHP is not ideal for domestic applications, due to engine sizing issues and would have a minimal effect on domestic carbon dioxide emissions and energy costs.

It has been demonstrated that a CHP/HP plant could effectively meet the need of a suburban dwelling and have significant financial and environmental benefits. The incorporation of a heat pump allows highly variable thermal and electrical demands to be satisfied. Additionally, surplus generating capacity is utilised and hence economic engine efficiencies are maintained. The advantages of domestic CHP/HP are evident from the significant reductions in domestic energy costs and reduced domestic carbon dioxide emissions.

The rating of a CHP/HP plant will depend on the demand profile of the host dwelling and the criteria on which the plant is optimised. A CHP/HP plant can be either optimised in environmental or economic terms. Environmental and economic plant optimisations are dependent on unrelated factors and hence plant configuration will differ.

The advantages of CHP/HP over conventional CHP has been clearly proven. Given the potential for domestic CHP/HP, a prototype plant was built to test the concept practically. The following chapters will detail the development of the prototype plant and associated modelling.

5. Prototype Plant Development

5.1 Introduction

The previous chapter introduced the CHP/HP concept: this chapter details the development of a laboratory based CHP/HP prototype plant. Aims and design philosophy of the prototype plant are discussed, before the development of individual sub-systems is covered. The structure of this chapter is shown in Figure 5.1, where the sub-systems are defined and the major components identified.

5.2 Aims of the Prototype

It was intended that the prototype development would be used to gain experience in practical domestic scale co-generation. Once the prototype was complete, it was used to investigate behaviour of a domestic scale CHP/HP plant and provide results for further analysis and comparison with modelled results. Additionally, the prototype plant development identified and solved practical problems that were not apparent from modelling exercises. Experience gained during the development and testing, with subsequent analysis, will be used to specify a second generation prototype plant.

5.3 Design Philosophy

For domestic scale CHP/HP to be commercially feasible, it must be demonstrated that the plant can be constructed with commercially available technology. To this end, most components were acquired from commercial sources. Some components were not available, and had to be purpose designed and built. Most of the commercially sourced components required some degree of modification, but this was kept to a minimum to meet the criteria of commercial feasibility. The design of the plant also had to be compact, to demonstrate the suitability of CHP/HP plant for a domestic installation.

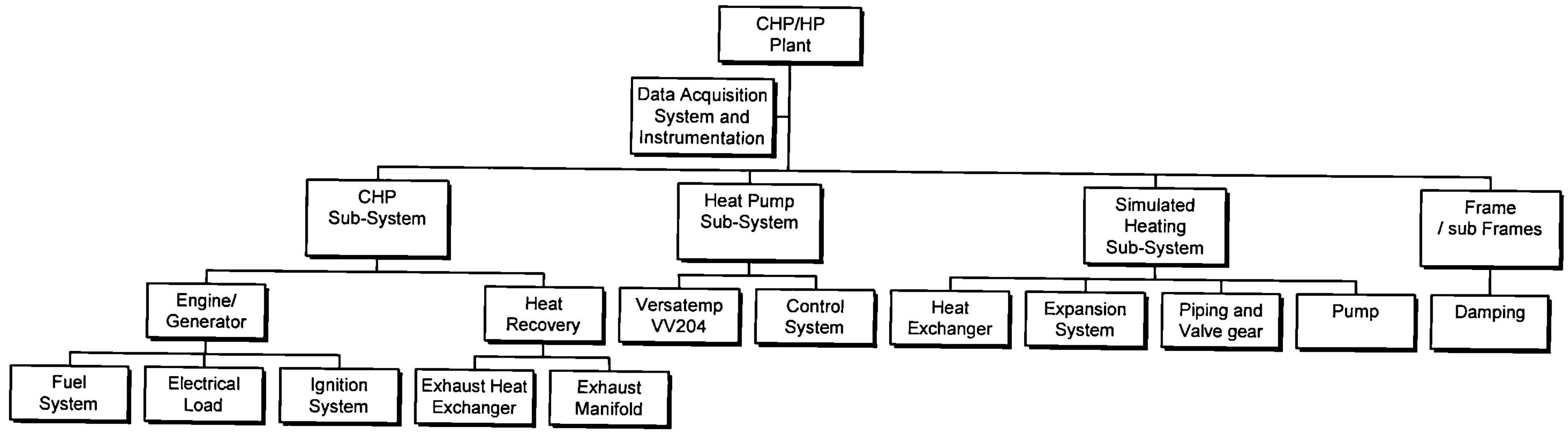

Figure 5.1 CHP/HP Plant Sub-Systems and Components

5.4 Plant Overview

Before the individual sub-systems and components are discussed in detail, it is necessary to give an overview of the plant. Figure 5.2 summarises the overall system layout. As the prototype plant was intended to demonstrate the suitability of CHP/HP to domestic applications, the heat delivery (or removal) of the plant must be via a low pressure water (LPW) system, as is found in domestic heating systems. The heat pump sub-system is used to pre-heat the LPW water flow prior to further heating in the CHP sub-system: Section 5.7 will discuss the LPW system in detail. A simulated heating load cools the return water, which is then re-circulated. The electrical output of the generator was either dissipated by a resistor bank or used by the heat pump, or both. If necessary, the heat pump could be powered with utility supplied electricity.

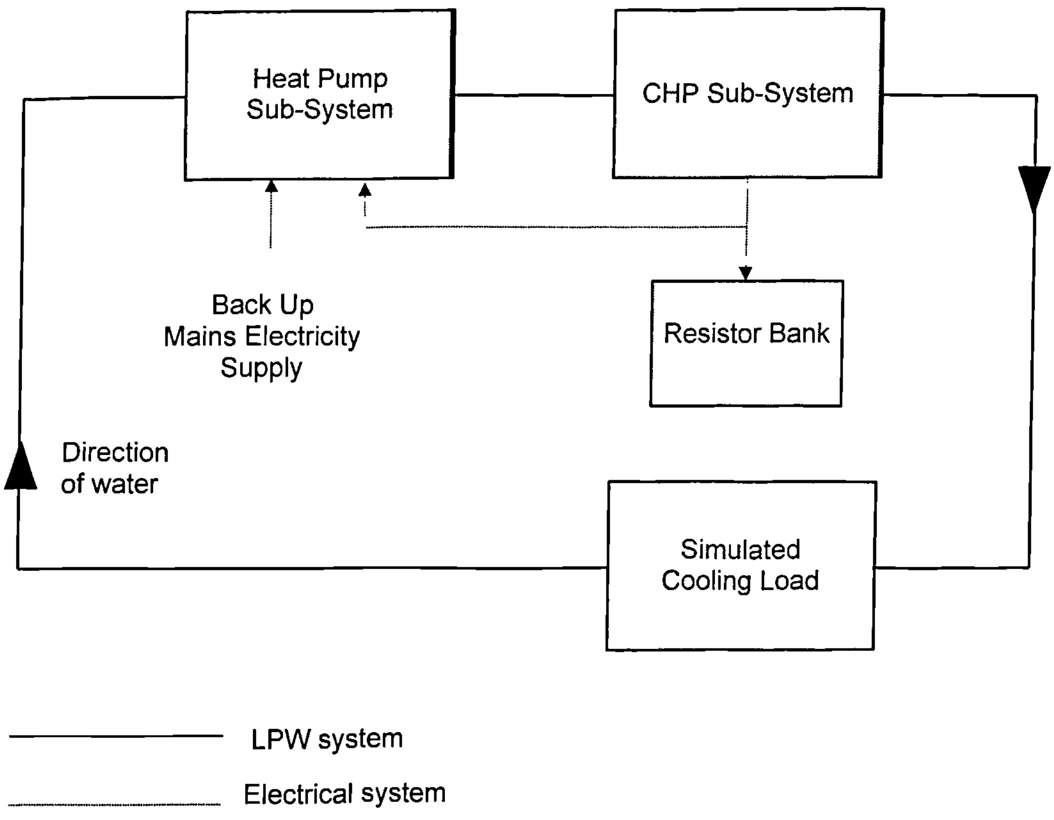

Figure 5.2 CHP/HP Plant Overview

5.5 CHP Sub-System

The CHP sub-system is essentially a conventional CHP plant, although there are some significant differences due to the reduced size. The development of the CHP sub-system is described in the following sections. Engine/generator set choice is summarised in Section 5.5.1 and modifications to the fuel system are described in Section 5.5.2. The development of the ignition timing system is discussed in Section 5.5.3 and the heat recovery/exhaust system will be considered separately in Section 5.5.4.

5.5.1 Engine/ Generator Choice

The choice of an engine is critical for a successful domestic CHP application. For a fully developed commercial plant, maintenance, noise/vibration and engine life would be major considerations. At this stage, it would be inappropriate to address the above issues in detail, as the research is intended to concentrate on the analysis of the overall CHP/HP concept, and the interaction of constituent components. Previous work has concentrated on the use of Stirling engines in domestic CHP applications (see Section 2.3.3). However, for the purposes of this work, the use of Stirling engines would detract research efforts away from the overall hybrid system. It was decided to use an inexpensive, commercially available engine/generator set as the basis of the CHP sub-system, for the following reasons:

- Inexpensive.
- Widely available engine and spares.
- Ease of maintenance.
- Ease of modification – fuel, ignition and exhaust systems required alteration.

Preliminary modelling demonstrates that an engine/generator set of between 1.1kWe and 1.2 kWe rating would be the optimum engine size to meet domestic energy needs. Given the criteria cited above, a number of commercial petrol four-stroke engine generator sets were examined for suitability. It was decided to use the Briggs and Stratton based IC3 unit, as it satisfied the stated criteria (see Appendix D.1.1 for full specification and description).

5.5.2 Fuel System

As the prototype plant intends to demonstrate a feasible domestic system, it was necessary to convert the engine to run on a mains gas supply. The conversion of a petrol engine to natural gas normally requires modification to the cylinder head and valve gear (see Section 3.2.2). Alterations were only made to the carburettor and ignition timing system (see Section 5.5.3), as extensive engine modification would not have contributed to the aims of the thesis.

Gas from the utility supply is first metered, then regulated to atmospheric pressure, before being mixed with air in the carburettor (see Figure 5.3). To comply with safety requirements, two gas isolation valves were fitted to the gas system on either side of the gas meter. Connection to the gas supply was via a domestic cooker 'quick release' fitting, so that the engine could be disconnected from the supply without infringing safety regulations. Given the small size of the engine, it was impractical to fit a gas diaphragm carburettor and hence an annulus was fitted on to the existing carburettor to inject gas into the air intake. This also allowed duel fuel use: see Figure 5.4. The air/fuel ratio was dictated by the diameter of the jet. By retaining the original carburettor, no modification to throttle/governor linkages was necessary.

In order to maintain the principle of sourcing components commercially, the conversion equipment was purchased from NGS Ltd (Birmingham). Bottled butane conversion equipment was modified, by reducing the regulator size. A full set of carburettor jets were manufactured to test different mixture settings (see appendix D for designs and relevant calculations).

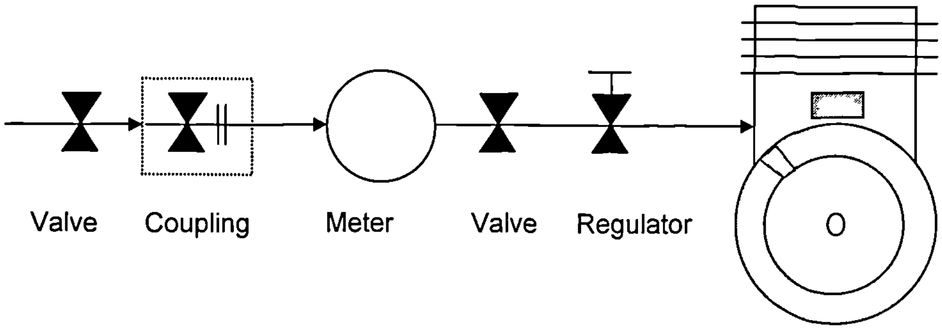

Figure 5.3 Fuel System

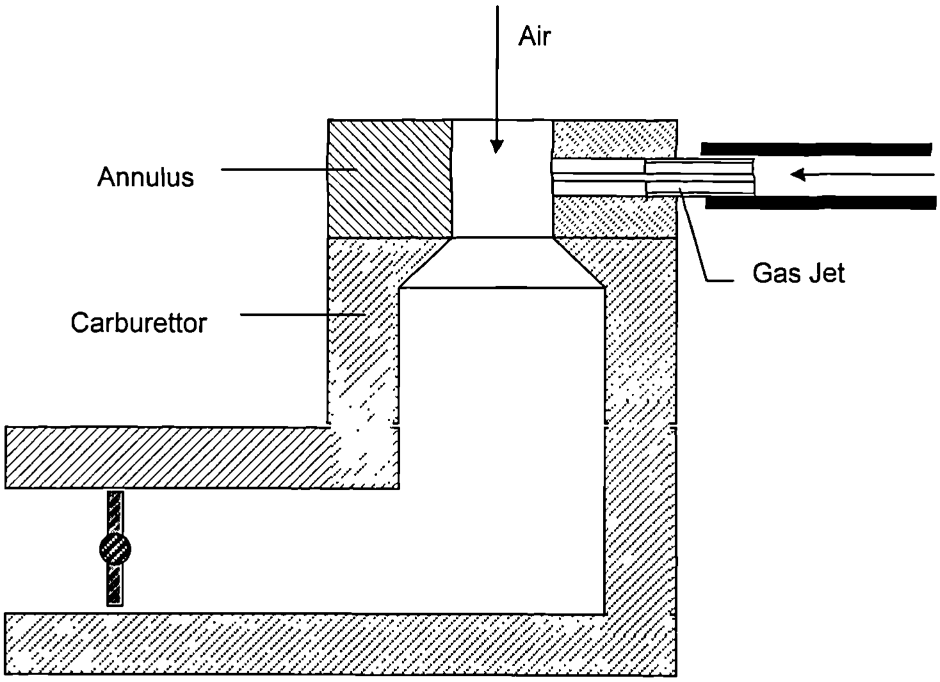

Figure 5.4 Carburettor Conversions

5.5.3 Ignition Timing System

The combustion characteristics of natural gas and petrol are different. Hence, as the IC3 engine used in this research was designed to run on petrol, it was necessary to optimise engine ignition timing.

The original ignition system comprised of a magnetic insert in the engine flywheel and a simple induction coil fixed to the cylinder block (see Figure 5.5). Due to the simple design, no adjustment was possible. The initial ignition system fired at 15^{O} top dead centre (TDC), to allow for a variation in fuel quality without pre-ignition occurring. Although satisfactory running could be achieved using the original ignition system, when running on natural gas, by retarding the ignition timing, greater performance of the engine could theoretically be achieved. As methane has a high octane rating, it would be possible to ignite the fuel at 0^{O} TDC, without pre-ignition problems, to maximise engine performance. An ignition system that could vary the ignition timing from -5^{O} to 15^{O} TDC was developed.

As the development of the electronic ignition system was relatively detailed, but did not contribute directly toward CHP/HP research, the work was carried out in conjunction with an undergraduate project under the supervision of the author.

The electronic ignition system developed (see Figure 5.6) was composed of:

- A transistorised magnetic sensor, mounted on a water cooled bracket above the fly wheel.
- An electronic control and conditioning unit, primarily designed and built by the undergraduate student.
- A high current transistorised switching unit (a standard automotive component).
- A standard automotive ignition coil.
- A standard spark plug.

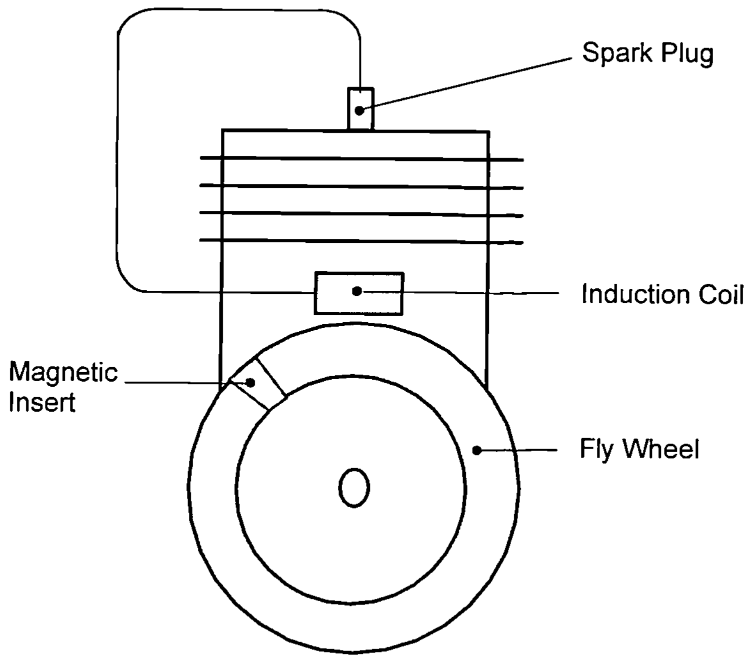

Figure 5.5 Standard Ignition System

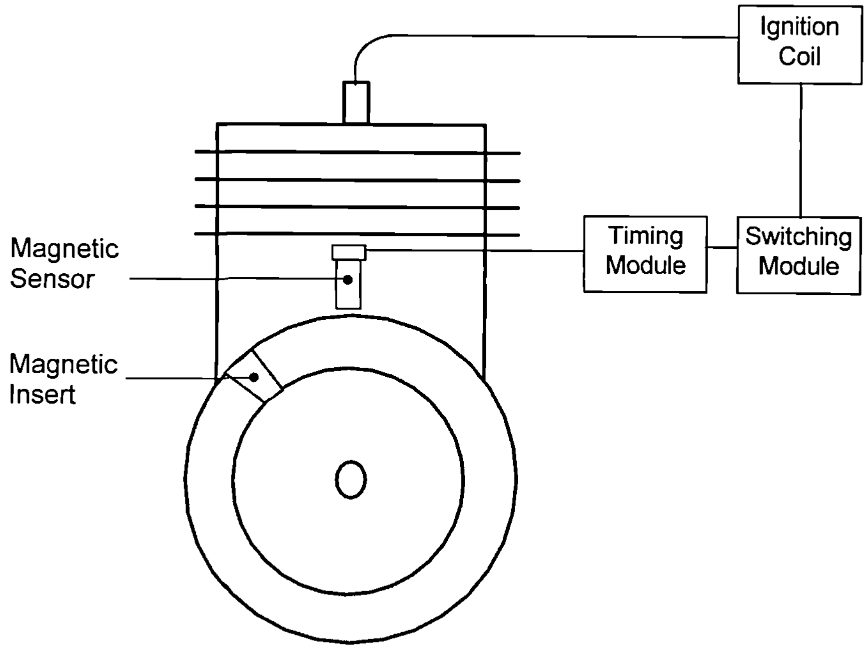

Figure 5.6 Electronic Ignition System

The operation of the ignition system (with reference to Figure 5.7) was as follows:

- As the magnetic flywheel insert passes the magnetic sensor, a voltage is generated.
- The signal from the magnetic sensor triggers a timer circuit. After a predetermined interval, the timing circuit produced an output signal. The output signal is in the form of a square wave. This is conditioned by an inverter and a Schmidt trigger to produce a sharp peak.
- The conditioned output signal causes the transistorised switch to operate, allowing current to briefly pass through the coil and produce a spark at the spark plug.

As the engine is at a fixed speed, it is possible to alter the ignition timing by varying the delay, providing the magnetic pick-up is mounted before -5^{O} TDC. The delay was varied by adjusting a variable resistor within the timing circuit.

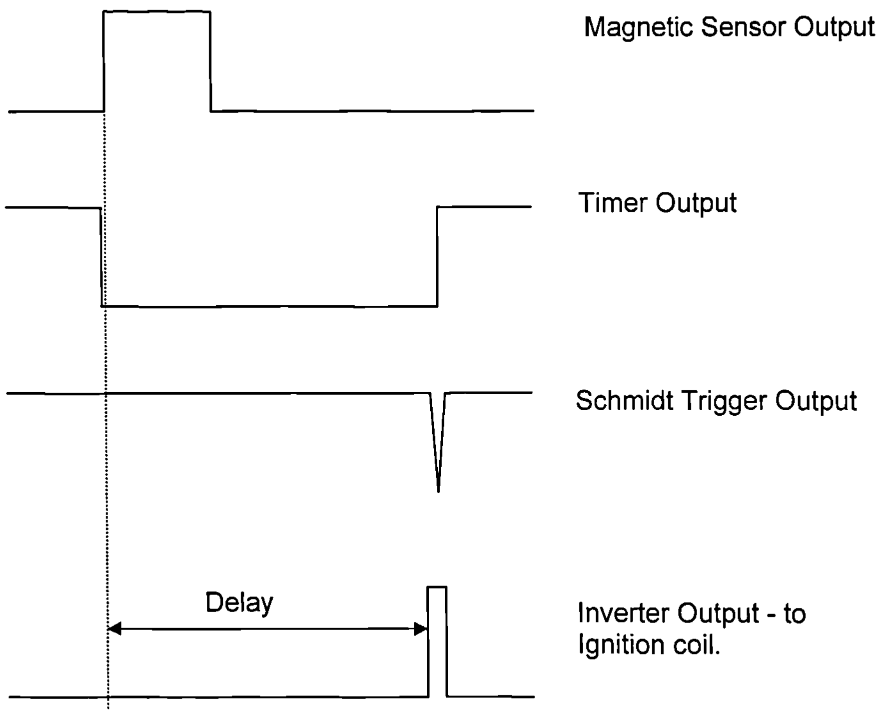

Figure 5.7 Electronic Ignition System Signal Conditioning

5.5.3.1 Ignition System Development

Initial testing of the electronic ignition system revealed a number of problems:

- *Multiple triggering* of the ignition system due to the di-pole nature of the magnetic insert. This led to erratic engine firing and increased engine vibration. This problem was overcome by adjusting the position of the sensor, increasing the delay period.

- *Electromagnetic noise* from the spark plug caused the timing module to trigger, producing a feed-back loop. As the spark plug was firing constantly, the mixture in the engine combusted prematurely and the engine stalled when placed on load. The problem was overcome by increasing the level of suppression employed.

- *Ignition coil overheating* occurred frequently. This was caused by impedance mismatches between the coil and the rest of the system. As ignition coils are designed for specific systems, a mismatch was inevitable. This problem was partially overcome by the addition of a large air-cooled ballast resistor. This allowed for over an hour of continuous running, which proved to be adequate for experimental purposes.

Development of the ignition system was discontinued once reliable performance was achieved. Although a number of refinements could have been made, they would have detracted from the core research. The use of the electronic ignition system in optimising engine performance is discussed at length by *Hand* [35].

5.5.4 Engine Heat Recovery System

This section chronicles the development of the engine's exhaust heat recovery system, which includes the exhaust manifold and engine heat exchanger (EHE) (see Figure 5.8). As the exhaust heat recovery is integral to the CHP sub-system, development effort was concentrated in this area. The majority of components had to be specifically designed and developed for the prototype plant, because of their small size, as no commercial components were available. These components were designed and built by the author due to the prohibitive cost of commercial manufacture of the specified items.

The final exhaust heat exchanger (EHE) design drew from the experience gained in the design, manufacture and assessment of two earlier models; a shell and tube design (designated EHE1 - see Section 5.5.4.1) and a multi-branch design (designated EHE2 - see Section 5.5.4.2). The final design, designated EHE3, is discussed in Section 5.5.4.3.

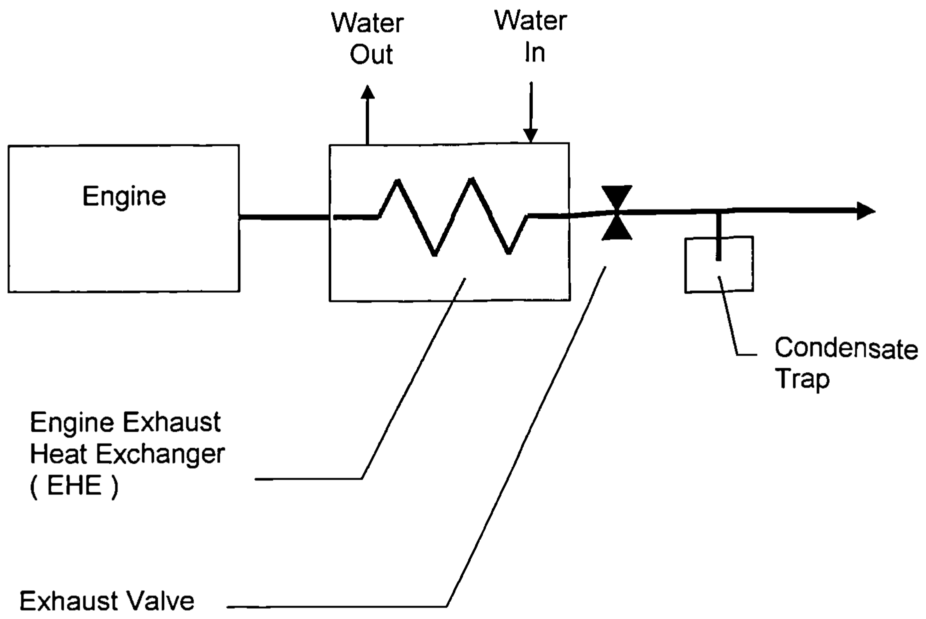

Figure 5.8 Engine Exhaust System

5.5.4.1 Shell and Tube Design – EHE1

Small-scale CHP heat recovery systems (see Section 2.3.2) were the basis of the initial exhaust heat exchanger (EHE1) design. This led to a simple counter-flow shell and tube arrangement being employed, where the exhaust gas flowed through the inner tube and heating system water flowed through the outer shell. Theoretical analysis was carried out to ascertain the optimal heat exchanger dimensions. The design was constrained by the engine exhaust port size, limiting the tube diameter to 17mm.

Figure 5.9 illustrates the EHE1 design. The construction was of welded mild steel tubing, with end caps welded to the tubing to complete the shell. Water connections were via conventional compression bulkhead fittings. Again drawing from small scale CHP practice, the heat exchanger was rigidly mounted on the engine (see Figure 5.9).

5.5.4.1.1 Assessment of EHE1 Design

Trials were carried out to assess the performance and practicality of the EHE1 design, from which the following observations were made:

- *Low heat transfer* - Heat recovery from the Mk1 design was poor due to a low heat exchanger effectiveness of 49% (see Appendix D.1.7).
- *Weld failure* - Rapid thermal cycling and a steep temperature gradient across the relatively short length of the EHE caused mechanical stress, ultimately leading to weld failure.
- *Fatigue of water connections* - As the heat exchanger was allowed to vibrate with the engine, the water hoses and connections experienced considerable fatigue which led to numerous leaks.

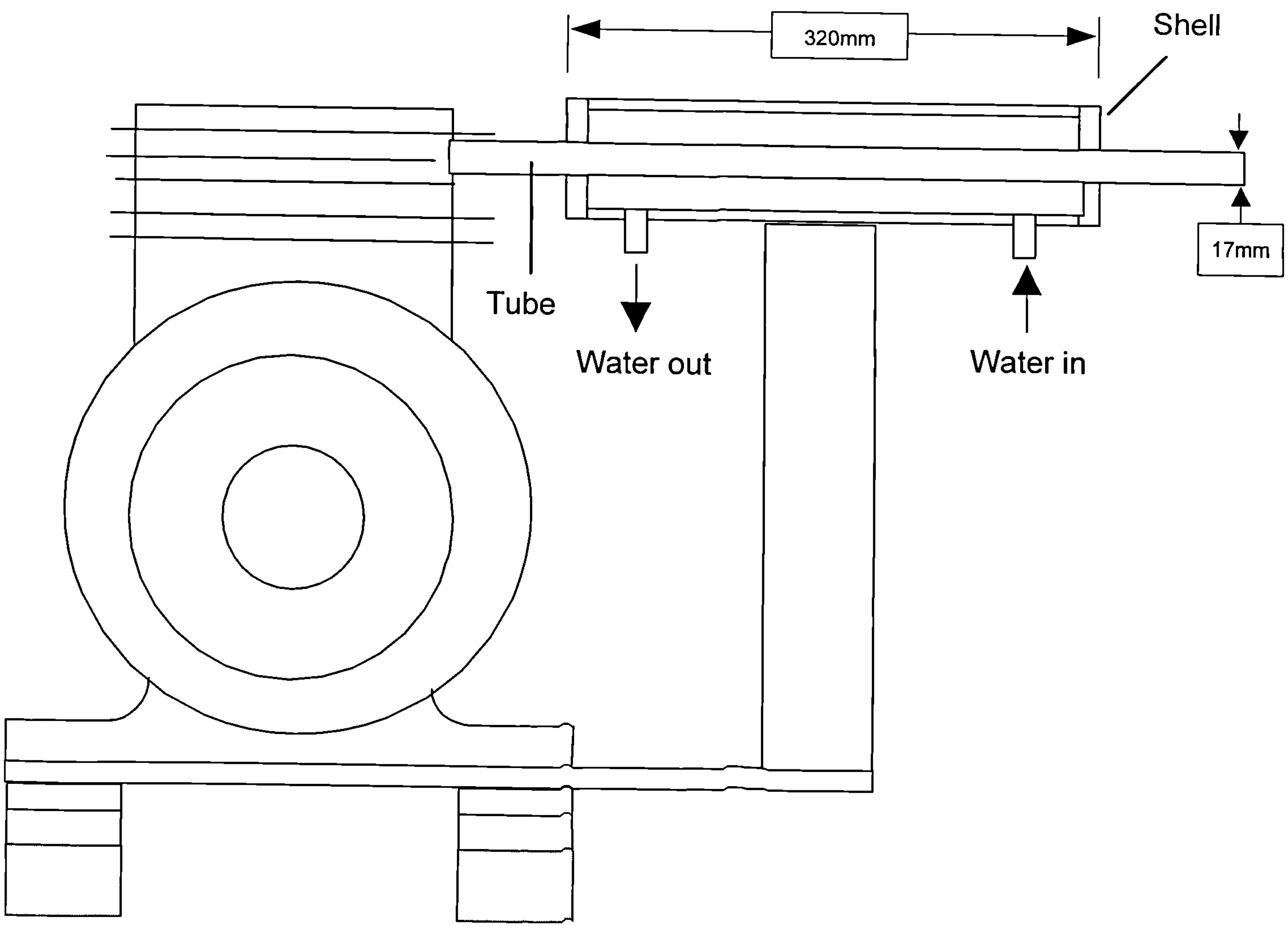

Figure 5.9 EHE1 Arrangement

5.5.4.2 Multi-Branch Heat Exchanger – EHE2

The second exhaust heat exchanger (EHE2) design addressed the problems experienced with the EHE1 design. In order to increase heat recovery, the heat exchange surface had to be increased. However, it was necessary to retain the compactness of the design. The results of EHE1 testing indicated that the surface area had to be increased by 600%, which would require a 2m long shell and tube heat exchanger - hence a new concept was required.

The approach taken was to use a multi-branch heat exchanger core to carry the exhaust gas, contained within a water jacket. Figure 5.10 illustrates the EHE2 multi-branch core arrangement. A multi-branch arrangement was chosen in preference to a multi-pass, as the parallel flow of exhaust gas would create less exhaust back-pressure than a series of connected loops.

New techniques had to be investigated in order to construct the core and water jacket. As the EHE2 design was considerably heavier than its predecessor, it had to be mounted separately from the engine. This required a flexible engine exhaust link (see Section 5.5.4.4)

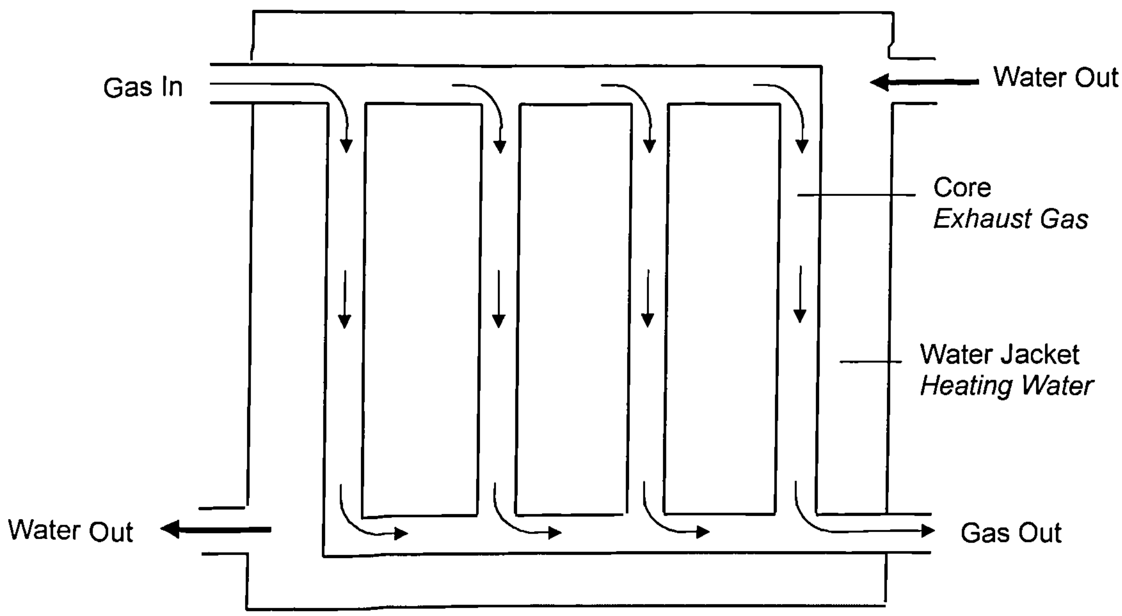

Figure 5.10 EHE2 Concept

5.5.4.2.1 Core Design

Constructing a multi-branch core presented complex jointing problems for the matrix of pipes required. Three types of construction were considered (see Table 5.1):

Table 5.1 EHE2 Core Manufacturing Options

Construction	Advantages	Disadvantages
Welded Mild Steel	Strong Vibration resistant.	Complex construction. Possible manufacturing errors.
Brazed Mild Steel	Strong Vibration resistant.	Complex construction. Possible manufacturing errors. Poor joint integrity at high temperatures.
Mechanical jointed Construction **(Compression fittings/ copper tubing).**	High thermal conductivity. Inexpensive. Corrosion resistant. Ease of manufacture. Ease of maintenance	Poor joint integrity with prolonged vibrations.

A core constructed from commercial plumbing compression fittings was the simplest and most adaptable option. The mechanically jointed core was intended to be used experimentally, to investigate the performance of the multi-branch approach, as it was thought that greater vibration resistance would be required in a final design. The use of brass/copper components, with high corrosion resistance, allowed for the condensation of exhaust gases, further increasing heat recovery. Figure D.4 (see Appendix D.1.9) illustrates the core design, which is doubled over to reduce space requirements. The engine manifold was also a brass compression fitting.

5.5.4.2.2 Water Jacket Construction

The water jacket design presented a number of problems, since it had to allow for the core accommodation, water and exhaust gas connections and ease of fitting/ removal.

Figure 5.11 below illustrates the basic form of the water jacket. The *main body* accommodates the core and the *end plate* completes the containment. A seal is made between the *main body* and *end plate* by a gasket made of relatively thick neoprene rubber, which deforms to overcome distortion on the sealing surfaces. The seal is completed by fixings and clamping strips. The main body was fabricated by welding pre-bent mild steel plates.

Water connections were made with commercial compression bulkhead fittings and sealed with neoprene rubber gaskets. The core outlet and inlet, sealed with a rubber and asbestos gasket respectively, also served as mountings for the core.

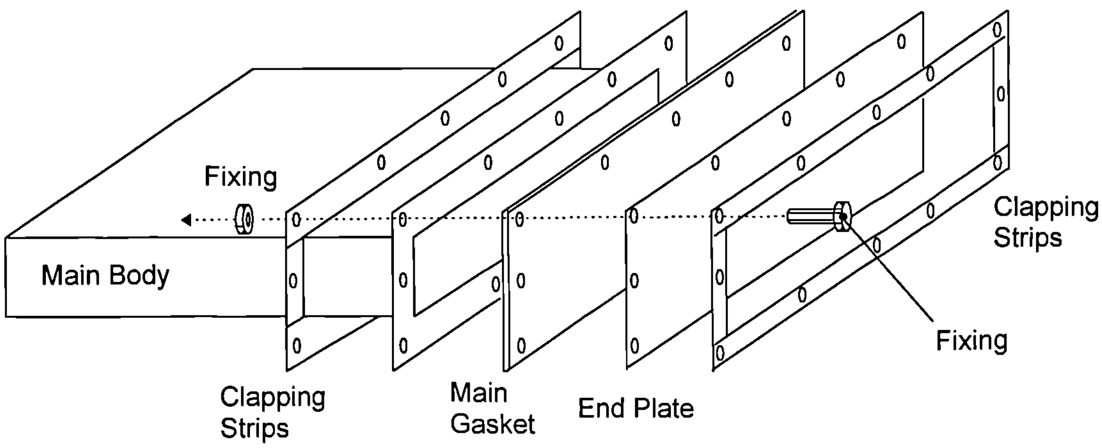

Figure 5.11 Exploded View of EHE2 Water Jacket Assembly

5.5.4.2.3 Assessment of EHE2

Initial testing of the EHE2 design resulted in the following findings:

- *Good Thermal Performance* - The thermal output of the EHE2 design was over $2.5kW_{th}$, with an associated effectiveness of over 80%, (see Appendix D.1.8).

- *Vibration Resistance* - The mechanically jointed core performed well, with no loosening of the compression fittings.

- *Distortion of Main Body* - The water pressure changes within the main body progressively deformed the sealing faces, ultimately leading to irreparable leaks.

5.5.4.3 Final Exhaust Heat Exchanger Design – EHE3

The final EHE3 design was of the same basic form as the EHE2 design, with inclusion of new features to overcome the highlighted problem. The thermal performance of the EHE2 design was satisfactory and the mechanically jointed core exhibited no indications of vibration damage, hence the core was retained and incorporated into the EHE3 design..

The water jacket was re-designed with the addition of strengthening members:

- The main body was contained within a welded angle iron frame to prevent distortion.

- The clapping strips were replaced with welded angle iron collars to prevent the sealing surfaces from distorting.

- The mild steel end plate was replaced with a heavy gauge aluminium plate.

The EHE3 design was included in the final prototype CHP/HP plant design. Figures D.5 and D.6 respectively show the front and side views of the EHE3 End Plate and sealing arrangement.

5.5.4.4 Flexible Exhaust Link and EHE Mounting

It was impractical to rigidly mount the EHE3 to the engine/generator, as in the EHE1 design, due to increased weight. The EHE3 was mounted directly on the *CHP sub chassis* (see Section 5.8.2). This was facilitated by the development of a flexible high temperature exhaust link. The flexible exhaust link was required to allow the engine to vibrate freely without transmitting movement to the EHE.

The flexible link is constituted of a length of stainless steel bellows welded to two end fittings (see Figure 5.12). The end fittings were fixed to the engine exhaust port and EHE exhaust inlet. This allowed the engine to vibrate freely as required, with minimal vibration transmitted to the EHE. An engine exhaust port extension was also required to create a gas-tight seal. Figure D.7 shows the Flexible Exhaust Link fitted to the EHE3 and engine.

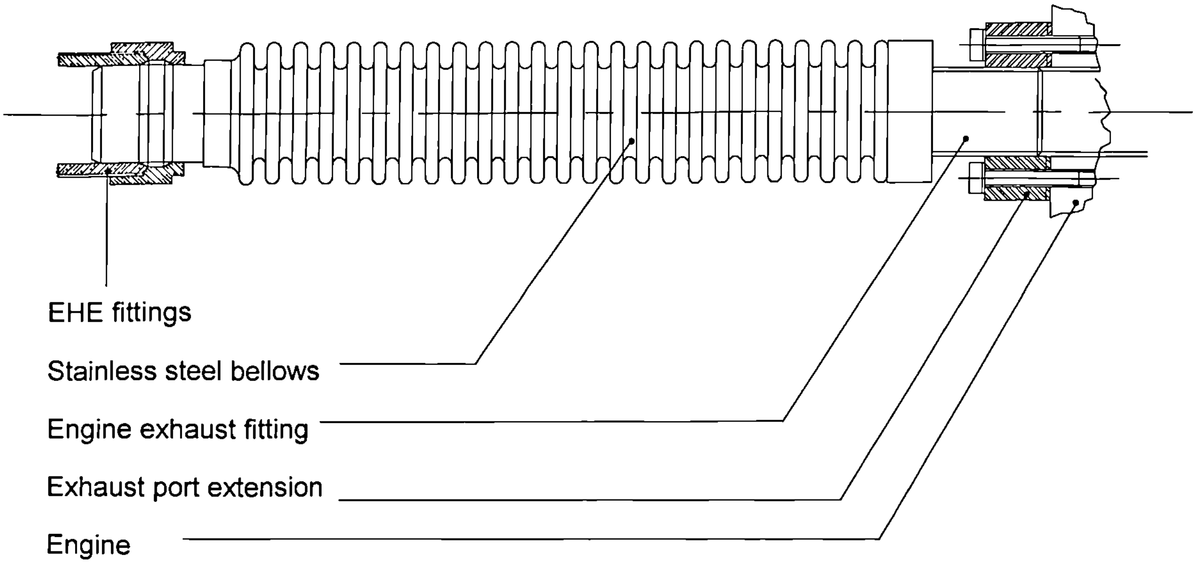

Figure 5.12 Flexible Exhaust Link

5.5.4.5 Exhaust System

The reminder of the exhaust system consisted of the following:

- *Exhaust Valve* - To prevent exhaust gas escaping when the CHP/HP plant was disconnected from the exhaust extract system, a 22mm brass gate valve was fitted after the EHE exhaust outlet. This also served as an exhaust break to shut down the engine if necessary.

- *Condensate Trap* - As the cooling of exhaust gas creates a large amount of condensate, it was necessary to fit a simple condensate trap before the exhaust gas entered the laboratory exhaust extract system.

5.6 Heat Pump Sub-System

Preliminary modelling identified the need for incorporation of a heat pump into a domestic co-generation plant (see Section 4.6). A commercially available air conditioning plant was used as the basis of the heat pump sub-system. The following section discusses and justifies the choice of heat pump and describes the chosen system.

5.6.1 Heat Pump Requirements

For a heat pump to be successfully incorporated into the prototype plant, a number of criteria had to be satisfied:

- Compatibility with heating water system.
- The ability to utilise engine cooling air.
- Compatibility with generator output.
- Reversible refrigerant system - to facilitate cooling mode (see Section 4.3.4).
- Compressor rating must match engine/generator rating (see Section 4.6.1).

Summarising the above criteria, an electrically driven heat pump with air/water heat transfer was required, with an electrical rating in the region of 1 to 1.5kWe, as shown in Figure 5.13.

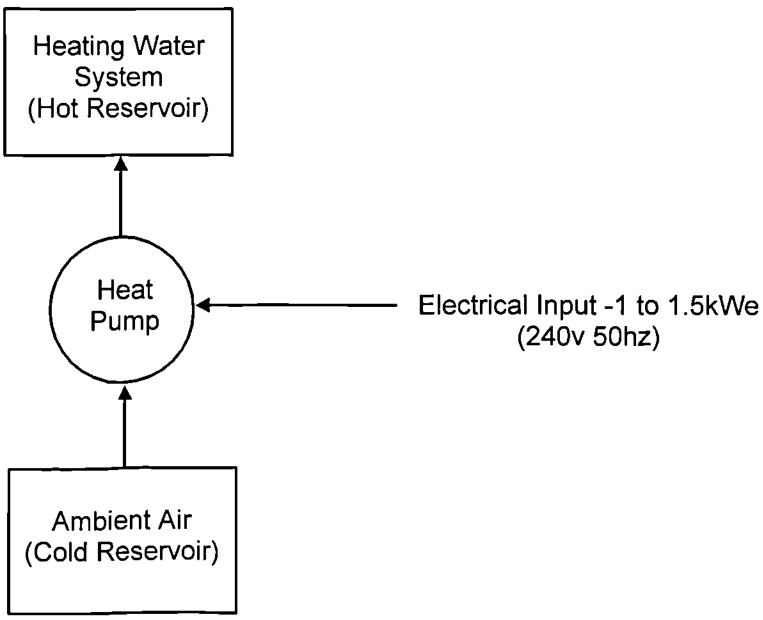

Figure 5.13 Required Heat Pump Specification

5.6.2 Heat Pump Choice

To comply with the design philosophy cited in Section 5.3, it was decided to purchase a commercial heat pump and to undertake appropriate modifications. Assembling a purposed designed heat pump from standard refrigerant components might have resulted in a more compatible design, but would have detracted from the main area of research.

The majority of commercial heat pump systems operate between air/air reservoirs, which reduced the range of possible suppliers. Two manufacturers of water/air heat pumps were approached:

- *Colerex* - Manufacturers of specialist heat pumps primarily aimed at the swimming pool market.
- *Temperature* - Manufacturers of water/air reversible heat pump systems to provide air conditioning for office accommodation.

The Versatemp VV 204 unit manufactured by *Temperature* was ideally suited for incorporation into the CHP/HP prototype plant, with an electrical rating of 1kWe and a reversible water/air heat transfer. Management at *Temperature* showed interest in the CHP/HP concept and as a result a *VV204* unit was purchased at a 50% discount. Technical support was also available at no charge.

5.6.3 Heat Pump Description

The main features of the VV204 heat pump unit (as shown in Figure 5.14) are discussed in this section, in the context of the prototype CHP/HP plant design, and with particular reference made to the interfacing of heat pumps with other components. Appendix D 1.4 gives further technical details of the VV204 unit.

5.6.3.1 Refrigerant System

The refrigerant system of the VV204 unit is based on a conventional vapour compression cycle, with the addition of a shuttle valve that allows for bi-directional flow of refrigerant - hence reversible operation can be achieved. Figure 5.15 illustrates the refrigerant system components, which are also described below.

Air Coil - The air coil is fabricated from an aluminium matrix through which the refrigerant flows. In the CHP/HP heating mode, the air coil forms the heat pump evaporator and the condenser in the CHP/HP cooling mode. Air flow through the air coil is assisted by a variable fan.

Water Coil - A copper coil carries the water supply, wrapped around a section of refrigerant pipe, facilitating water/refrigerant heat transfer. The water coil constitutes the heat pump evaporator in the CHP/HP cooling mode and condenser in the heating mode. The optimal water flow rate is 0.00663kg/s. Owing to the refrigerant used, the operational water temperature must be in the range of 20^{O}C to 43^{O}C.

Compressor - The heat pump compressor is electrically driven, hermetically sealed and requires a start up current of 20Amps.

Expansion Device - refrigerant expansion is regulated by capillary tubes.

Reversing Valve - The valve used to reverse the refrigerant design is electrically operated by the on-board control system, via a solenoid.

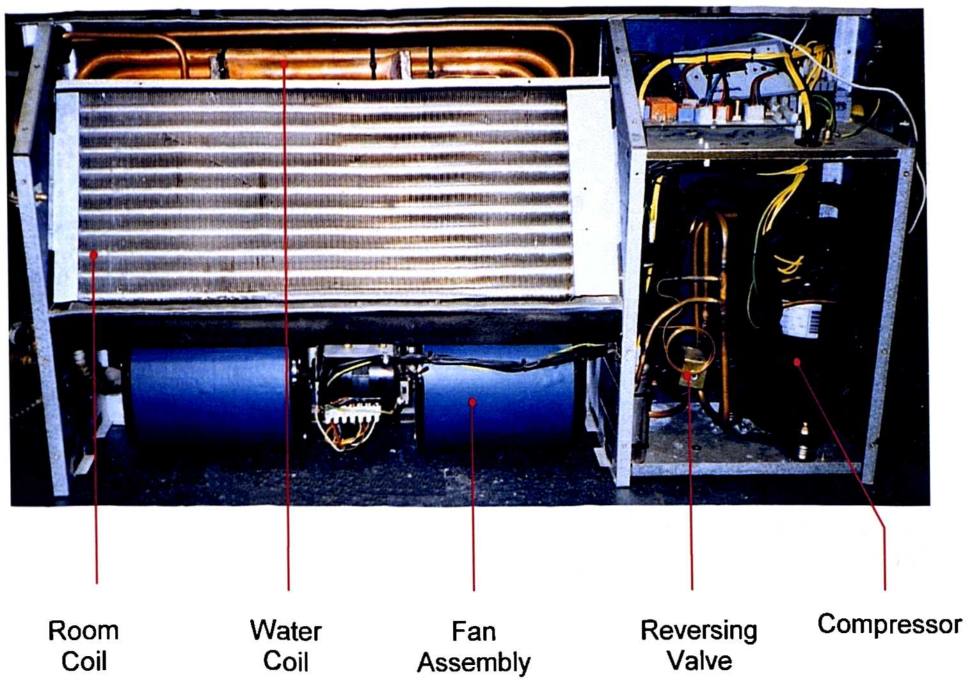

Figure 5.14 Versatemp VV204 Unit

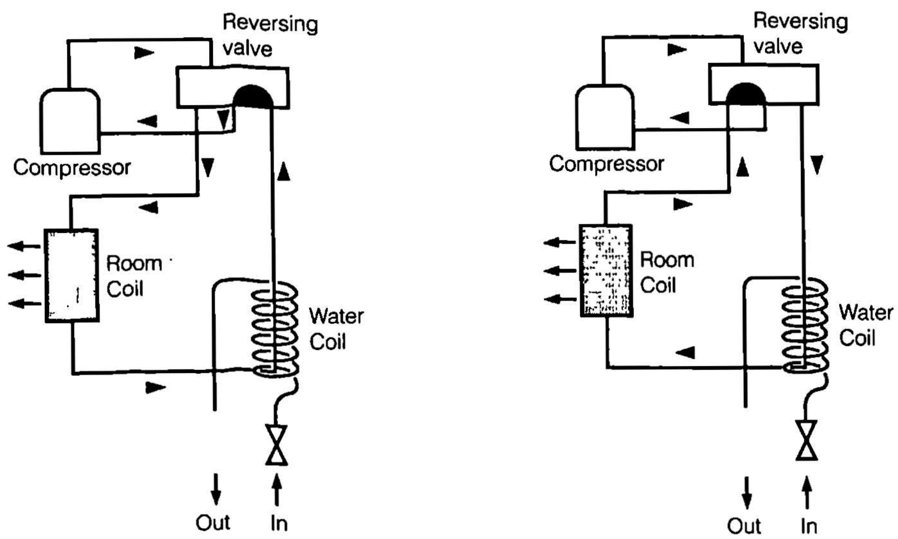

Figure 5.15 Versatemp VV204 Refrigerant System

5.6.3.2 Control System

The VV204 heat pump has a sophisticated on-board microprocessor control system, including a user control panel and associated transducers. The manual settings control:

- Air coil fan speed.
- Air outlet set point temperature. As the VV204 is designed as an air conditioning unit, control is referenced to air temperature, which gives control of water temperature. The desired air temperature is achieved by the modulation of the refrigerant system.

Transducers fitted to the control system measure:

- Water outlet/ inlet temperatures.
- Air inlet temperature.
- Refrigerant pressure.

The control system also protects the refrigerant system from extremes of temperature and pressure. If an out of range value is reached, then the control system will disconnect the compressor power supply until the parameters have returned to a safe value.

5.6.4 Heat Pump Modifications

Modifications to the VV204 heat pump unit were kept to a minimum, as the unit formed a discrete and complete sub-system. Minor modifications were made for instrumentation purposes. The control system remained separate from the general plant instrumentation to avoid potential errors.

5.7 Low Pressure Water System

This section describes the low pressure water system (LPW system) of the prototype CHP/HP plant and the simulated heating load (see Section 5.7.1). As explained in Section 5.4, the LPW system was heated (or cooled) by the heat pump and CHP sub-system, and then cooled by a heating load - intended to simulate a dwelling's heating demand. The heat recovery aspects of the heat pump and CHP sub-system have been covered in previous sections and discussion will be limited to aspects of the LPW system not referred to in other sections. To complement the overview of the LPW system given in Section 5.4, a more detailed explanation of the operation of the system (see Figure 5.16) is given below:

- Cooled return water enters the heat pump sub-system and is pre-heated by the heat pump (the heat pump sub-system is by-passed when the plant is operating in the CHP mode). The return water was intended to range from 20^{O}C to 40^{O}C, and optimised at 33^{O}C.
- The preheated water enters the CHP sub-system and is heated further by the CHP plant exhaust heat exchanger. The outlet water temperature was intended to be in the region of 70^{O}C.
- The heated water is cooled in the simulated heat load to the 20^{O}C to 40^{O}C range.
- A pressurisation unit maintains a pressure head on the system and allows for expansion.
- The water is returned to the heat pump by the circulation pump, which is throttled to obtain a flow rate of 0.06kg/s (see Section 5.6.3.1).

The return and flow temperatures, water pressures and flow rates approximate those found in domestic heating systems. Figure 5.16 overleaf is a detailed schematic of the LPW system. The Appendix D.1.2 reviews relevant details of the LPW system and components. Instrumentation fitted to the LPW system will be discussed in Section 5.9.

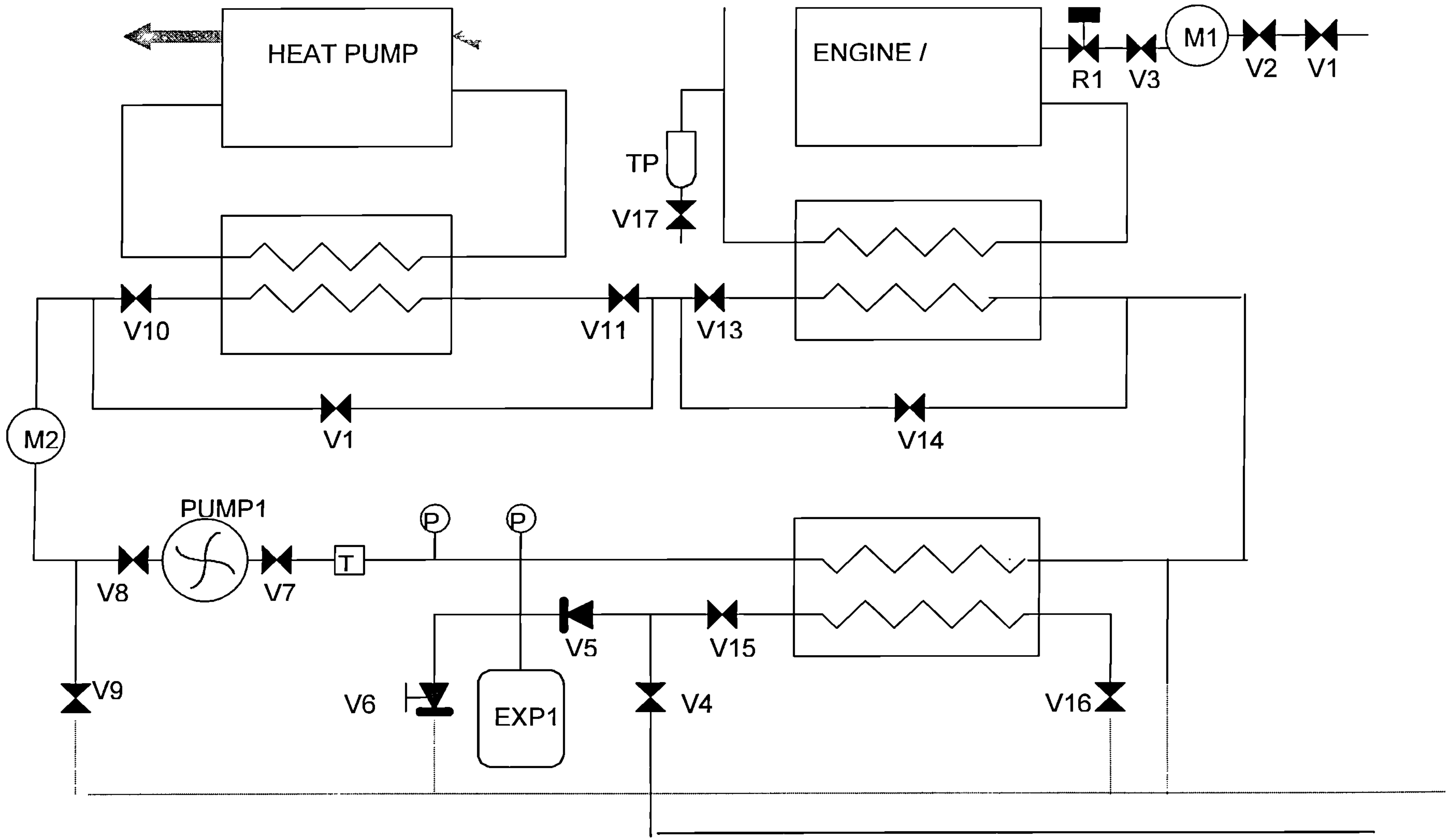

Figure 5.16 LPW System (see Appendix D.1.2 for component details)

5.7.1 Simulated Heating Load

The simulated heating load was required to mimic the heating load of a dwelling. This was achieved by the manufacture of a water/water heat exchanger (which drew from the experience gained in the manufacture of EHE2/EHE3 designs). The LPW system flowed through a central core and was cooled by a secondary water flow within the surrounding jacket (see Figure 5.17).

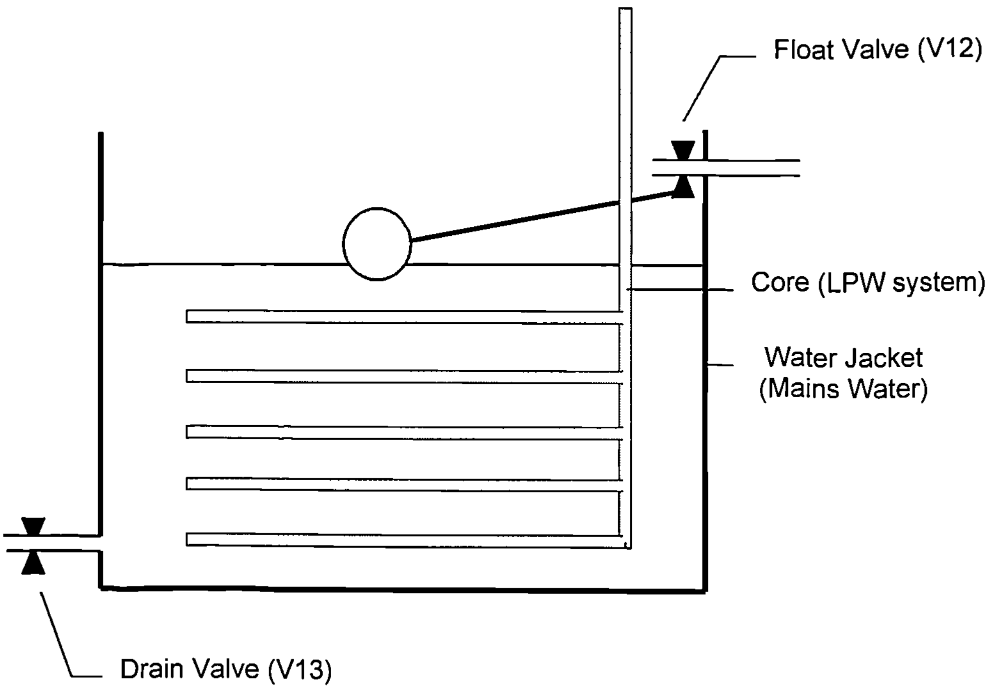

Figure 5.17 Simulated Heat Load Assembly

Water Jacket - The water jacket assembly consisted of a commercial 5 gallon plastic water tank with a drain and level valve fitted. The mains water supply was connected to the level valve. The rate of water flow (hence rate of heat transfer) was controlled by throttling the drain valve. As the water level dropped, the float valve adjusted to give a constant water level.

Core - The copper core that carried the LPW system was fabricated from commercial copper tubing with soldered joints. To maximise use of the available area, the core was a multi-branch/ multi-pass design, forming a cubic matrix.

5.8 Plant Chassis and Vibration Suppression

The following section briefly discusses the structure of the CHP/HP prototype plant chassis and the attenuation of engine vibrations.

5.8.1 Main Chassis

In order to retain the compactness of the prototype plant, the majority of the sub-systems and components were accommodated on the main chassis. The LPW system was mounted to the main chassis along with the CHP sub-chassis (see Section 5.8.2). The heat pump sub-system was a free standing unit that did not require mounting on the main chassis. Figure 5.18 shows the completed prototype plant assembly fitted to the main chassis. Appendix D.3 contains additional Figures of the completed assembly.

5.8.2 CHP Sub-Chassis

The CHP sub-system, including engine/generator set and EHE, was mounted on an individual sub-chassis, which was mounted on the main chassis. By mounting the CHP sub-system in this manner, it could be removed and refitted without disturbing any other components except for water connections. The CHP sub-chassis was fabricated from 50mm RHS x 50.

5.8.3 Vibration Attenuation

The single cylinder design of the IC3 engine (see Section 5.5.1) used in the prototype gives rise to severe vibration, which must be attenuated to prevent damage to other components. After consultation with *Dunlop-Metlastic,* the engine/generator set was mounted on double 'U' mounts. Figure 5.19 illustrates the mounting arrangement. The engine/generator set is fixed by two transverse bars, which are supported by four double 'U' mountings, above the CHP sub-chassis (see Figures 5.19 and D.8). This allows the engine to vibrate laterally while isolating the CHP sub-chassis.

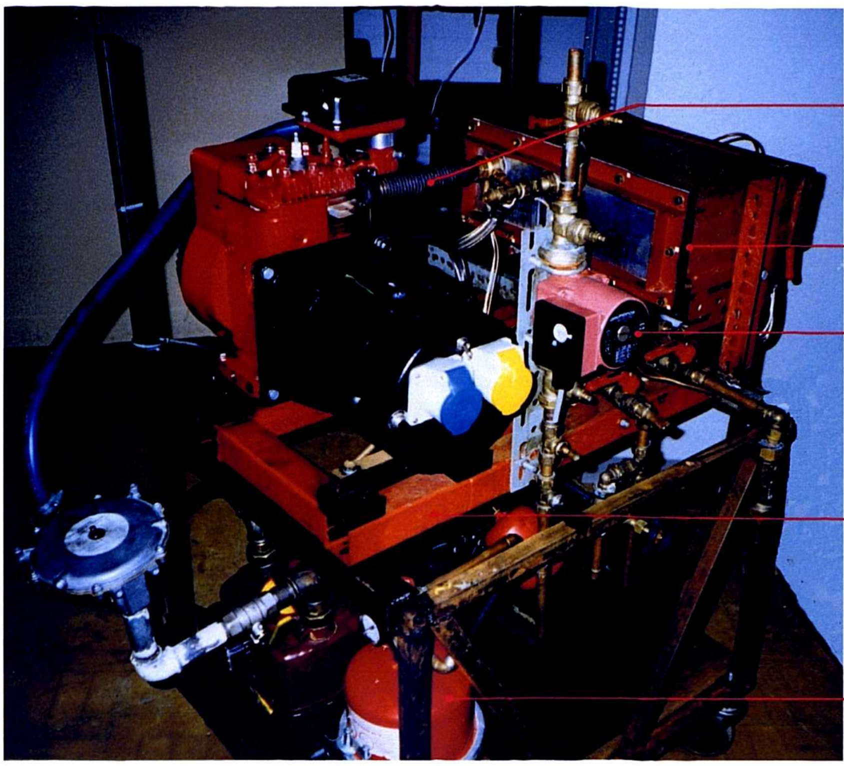

Figure 5.18 Completed Prototype Plant Assembly fitted to Main Chassis

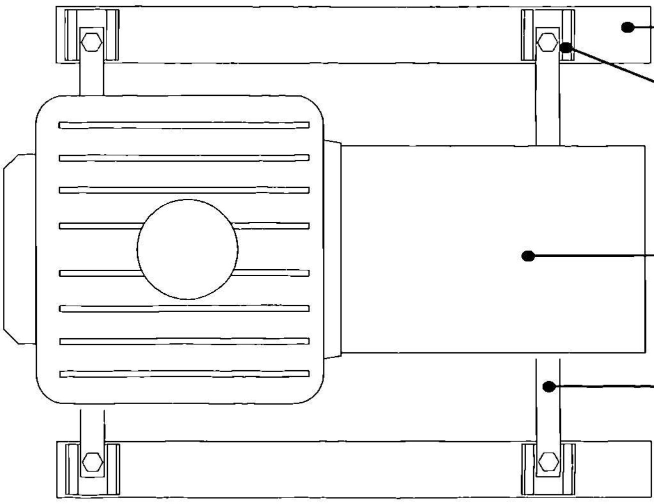

Figure 5.19 Engine/Generator Mounting

5.9 Instrumentation and Data Acquisition

The prototype CHP/HP plant was monitored by a computer based data acquisition system running specifically written software. The computer acquisition system hardware and software will be reviewed in this section. Details of transducer design and calibration are contained in Appendices D.1.3 and D.2 respectively.

5.9.1 Computer Data Acquisition System Hardware

The computer data acquisition system hardware consisted of an IBM compatible PC, a multi-channel analogue to digital converter (ADC) and a connection board (see Figure 5.20). The ADC used was a Keithly-1800 model set up to read differential analogue signals on 32 channels.

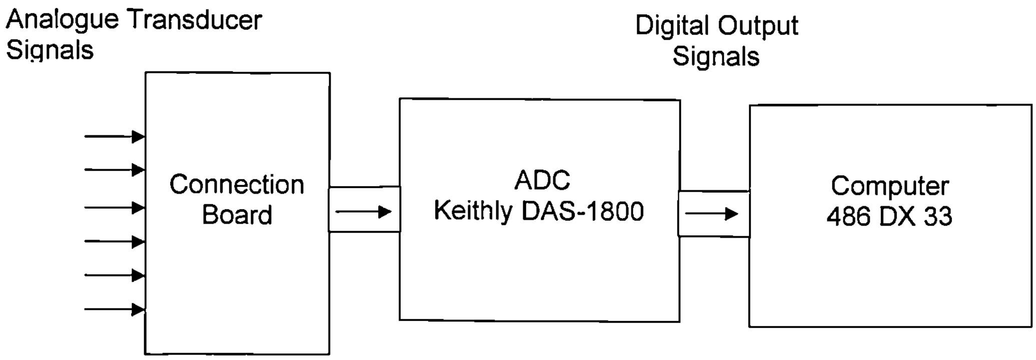

Figure 5.20 Instrumentation System

5.9.2 Software

To calibrate, analyse and store the data from the transducers, software was written using the *TestPoint* compiler (by *Keithly)*. The purpose written software converted the raw signal from the transducers, according to each transducer's calibration (see Section D.2). The results were displayed on the program's interface and stored every two seconds. The converted signals were further averaged every 30 readings to filter any errors.

5.9.3 Sensor and Transducers

The transducers used to monitor the prototype plant are broken down in related groups according to function in Figure 5.21 and are discussed in detail in Appendix D.1.3. Table 5.2 identifies the measured plant parameters and Figure 5.21 indicates each transducer's physical location.

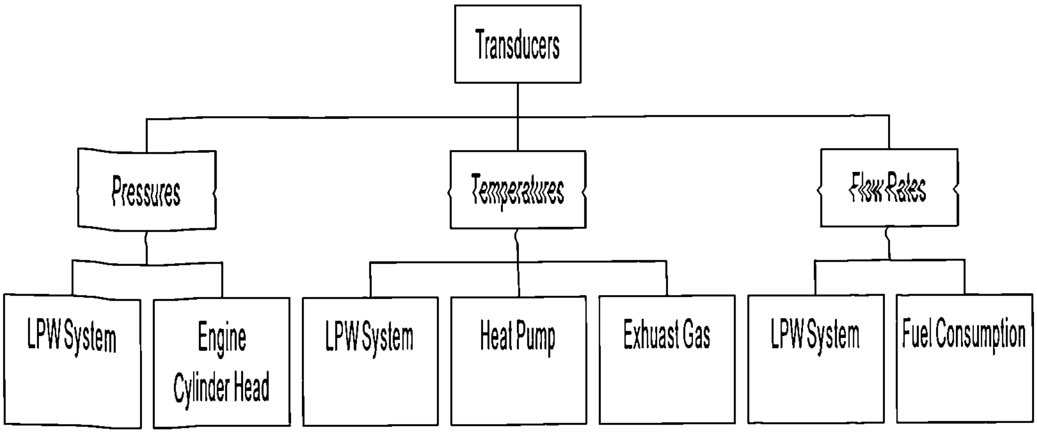

Figure 5.21 Transducer Location

Table 5.2 Transducer Descriptions

Parameter	Symbol	Range	Sensor
LPW System return temperature	T_1	10^oC -110^oC	LM35Z
LPW Heat Pump Water inlet temperature	T_2	*10^oC -110^oC*	*LM35Z*
LPW Heat Pump Water inlet temperature	T_3	10^oC -110^oC	LM35Z
LPW - EHE inlet temperature	T_4	10^oC -110^oC	LM35Z
LPW - EHE outlet temperature	T_5	10^oC -110^oC	LM35Z
EHE exhaust inlet temperature	T_6	10^oC - 450^oC	PT100
EHE exhaust outlet temperature	T_7	10^oC -110^oC	LM35Z
Heat pump Air Inlet/ Refrigerant temperature	T_8	10^oC -110^oC	LM35Z
Heat pump Air Outlet/ Refrigerant temperature	T_9	10^oC -110^oC	LM35Z
Heat Pump Compressor Temperature	T_{10}	10^oC -110^oC	LM35Z
LPW system pressure	P_1	0 - 3 bar	gwd1
Engine cylinder head Pressure	P_{eng}	0 - 100bar	Check JM
Fuel Consumption	m_f	0 - Check	GBm1
LPW system flow rate	m_{wt}	0 - 0.1lt/s	Turbo
Heat Pump Voltage	V_{hp}	200v - 250v	Hall effect
Heat Pump Current	I_{hp}	0 - 5A	Hall effect
Generator Voltage	V_{gen}	200v - 250v	Hall effect
Generator Current	I_{gen}	0 - 5A	Hall effect

6. Prototype Plant Testing and Analysis

The following chapter describes the commissioning, experimental testing and subsequent analysis of the prototype plant. Initially, the commissioning exercise will be described. General performance characteristics of the commissioned plant will be examined, prior to the presentation and analysis of overall steady state plant performance. First law and second law (exergy) analysis will be carried out separately. Economic analysis of the prototype plant operation will be carried out in Chapter 8.

Experimental data and analysis will be used in the following chapter to construct a model of the prototype plant (a *virtual plant*) so that plant performance can be extrapolated and further analysed (see Chapters 7 and 8).

6.1 Commisioning and Procedural Plant Testing

Plant commissioning was necessary to identify any problems with components and to optimise the performance of individual sub-systems. Although this stage of testing does not directly provide data on the CHP/HP concept, much useful experience was gained relevant to future domestic scale co-generation work.

6.1.1 Practical Problems Encountered

Section 6.1.1 details problems encountered in the commissioning and initial running of the plant and subsequent solutions.

6.1.1.1 Condition of Generator Electrical Output

The condition of the generator AC electrical output was such that it was incompatible with the heat pump on two counts.

- Unclean AC wave form.
- Generator governor response.

Given the single cylinder design of the engine/generator set, the waveform of the generator output is non-sinusoidal in nature. The waveform also varies with engine load, with features becoming more pronounced with greater engine load. The micro-electronic controller of the heat pump required a 'clean' sinusoidal AC supply, as with most micro-electronics. This gave rise to intermittent operation of the control system, prior to any attempt being made to drive the heat pump. This suggested that a multi-cylinder engine or DC generator ideally should have been employed: this will be discussed in Section 9.4.2.

A further problem experienced as a consequence of generator output by the heat pump was the response of the engine/generator governor to a change in load. When the heat pump compressor was operated, the engine speed dropped due to the increased load and then increased under the action of the governor. The variation in engine speed and hence the frequency of generator output, caused catastrophic damage to the heat pump controller, which required replacement.

As a consequence of both the above effects, electrical output of the generator was unsuitable to drive the heat pump. To overcome this would require extensive modification to both generator and heat pump. To prevent further damage, the generator dissipated all electrical output in a resistor bank, while the heat pump was powered by the mains supply. Although the actual electricity generated by the plant was not utilised by the heat pump, the generator output was set so that the equivalent electrical energy would be dissipated in the resistor bank. The effect is to simulate the operation of the envisaged electrical system. Figure 6.1 illustrates the resistor bank arrangement, where the electrical delivery of the engine/generator set (w_e) is equivalent to the heat pump compressor demand (w_{hp}) and simulated electrical demand (D_e).

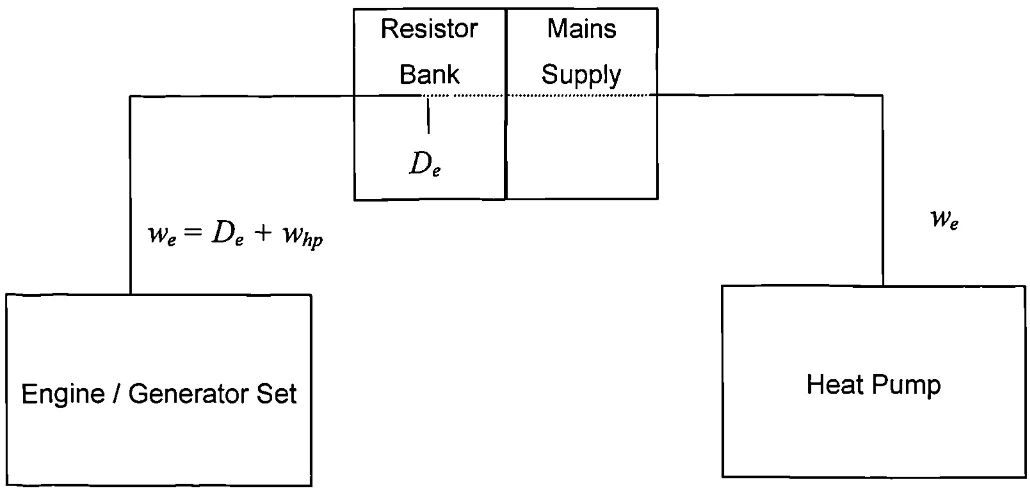

Figure 6.1 Equivalent Electrical System

6.1.1.2 Exhaust Back Pressure

Exhaust back-pressure within the laboratory exhaust ducting had a progressive effect on engine performance. Figure 6.2 shows the back pressure on the EHE exhaust outlet against time. It can be seen that the relative pressure increase over the EHE is small (owing to the design). An increase in back pressure was observed in the exhaust ducting after 30 minutes, accompanied by large sporadic variations in the pressure (not shown in Figure 6.2). As a result, the engine could be run for a 35 to 45 minute period before the back pressure caused detrimental effects.

It was concluded that some water vapour in the engine exhaust was condensing after the condensate trap and in addition some condensate was not trapped. From calculation, 0.46kg/s of water was produced by the engine at full load[1] and tests showed that the trap collected approximately 0.25kg/hour of condensate . The remaining 0.2kg/s of condensate built up in the exhaust duct, generally not affecting back pressure, until a sufficient quantity had accumulated to give a sharp rise in back pressure. The effect of wind and climatic conditions would have had an effect on the back pressure. However, draining of exhaust ducting demonstrated condensate build up was the predominant influence.

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[1] By applying the constant calculated in Appendix C.4 to a fuel consumption of 7kW.

This problem could not be satisfactorily overcome, as it was the consequence of the *artificial* laboratory environment, which had an extensive ducting system. A plant within a suburban house would not be subject to such an effect. However, the plant could reach thermal equilibrium within the operating time period (see Section 7.5.3.2 and Figure 7.19). Thus experimental results were obtained within the stated period. After each test the ducting was opened and purged of any condensate.

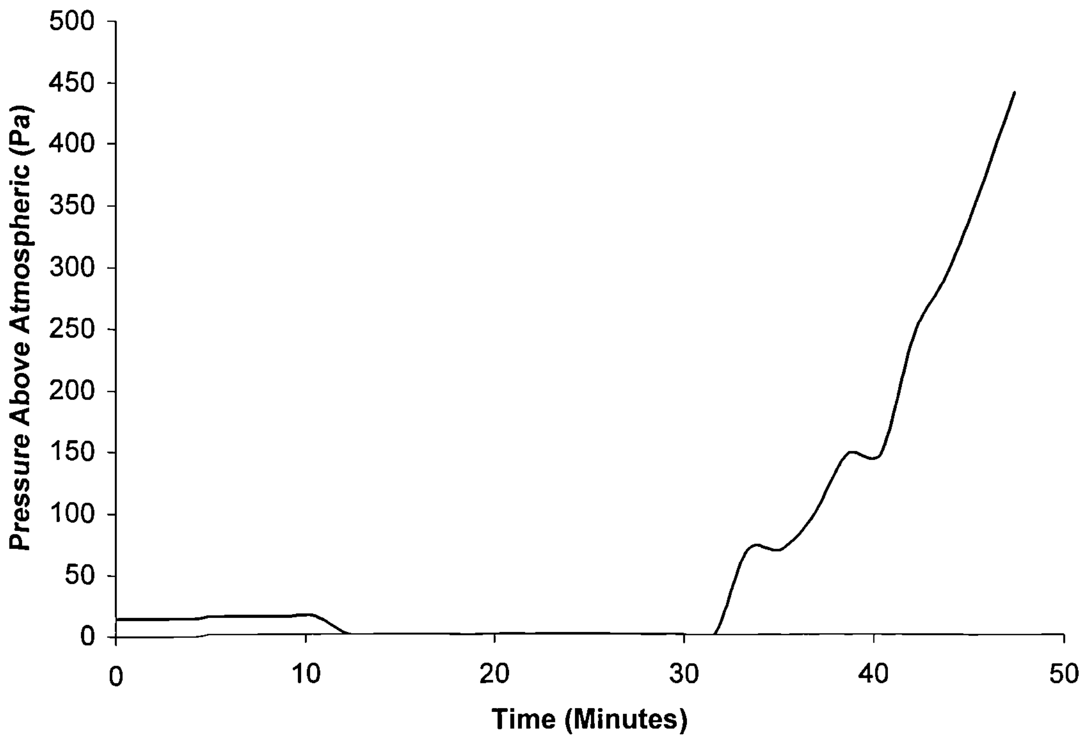


Figure 6.2 Back Pressure against Time

6.1.2 Plant Optimisation

Before testing the plant, some component optimisation was undertaken. The following parameters were optimised (other parameters were fixed by manufacturers):

- Air/fuel ratio (see Section 6.1.2.1).
- Ignition timing [35].
- LPW system flow rate (as per Section 5.6.3.1).

6.1.2.1 Air/Fuel Ratio

The air/fuel ratio was set by the diameter of a jet within the carburettor (see Section 5.5.2 and Appendix D.1.6). A number of different jet diameters were experimentally tested. Table 6.1 shows the air/fuel ratio with respect to engine/generator fuel conversion efficiency (η_e). A jet diameter of 3.5mm (which provided an air/fuel ratio of 1:11.6) was chosen as it returned the highest efficiency. This jet size also gave the most reliable engine starting. Comments on engine efficiency will be made in Section 6.2.1.

Table 6.1 Carburettor Jet Performance

Jet	Efficiency
mm	%
3.20	13.3
3.50	13.8
3.75	13.0
4.00	13.2

6.1.2.2 Miscellaneous Modification and Practices

Experience gained in the commissioning phase led to the following practices being adopted to maintain best engine performance. The engine generator set was stripped down and carefully reassembled so that bearings in the generator could be realigned, since a mis-aligned bearing was the source of mechanical losses. The engine cylinder head was regularly 'de-coked' and new cylinder head gaskets fitted. The spark plug was regularly cleaned and electrode gap checked.

6.2 General Plant Performance Characteristics

The following section will present the general performance characteristics of the commissioned plant. Results contained in this section will show performance characteristics with respect to varying conditions or transient effects. These are:

- Engine/generator electrical efficiency with respect to engine load.
- Steady state EHE thermal delivery with respect to engine load.
- Heat pump COP with respect to heat pump mechanical input.
- Thermal transient effects of HP compressor on COP.

All the following results were obtained during specific experimental tests. As these characteristics are used in the concept evaluation model (developed in Chapter 7), empirically derived functions representing the characteristics will also be presented. Additionally, the statistical validity of the functions will be discussed.

6.2.1 Engine/ Generator Electrical Efficiency

The relationship between engine/ generator electrical efficiency (η_e) and engine load (w_e) is shown in Figure 6.3, where:

$$\eta_e = \frac{w_e}{Q_f} \qquad (3.4)$$

With reference to Appendix E.1.1, this characteristic was obtained by incrementally increasing the engine load and measuring the fuel input.

The experimentally obtained function (presented in Figure 6.3) describes an approximate exponential curve - as would be expected. The r^2 value obtained indicated that the derived function is statistically reliable. The maximum efficiency obtained (approximately 16%), is relatively low for a natural gas fuelled engine, when compared with the Queens Building CHP plant at an electrical conversion efficiency of 29% (see Section 3.3.2).

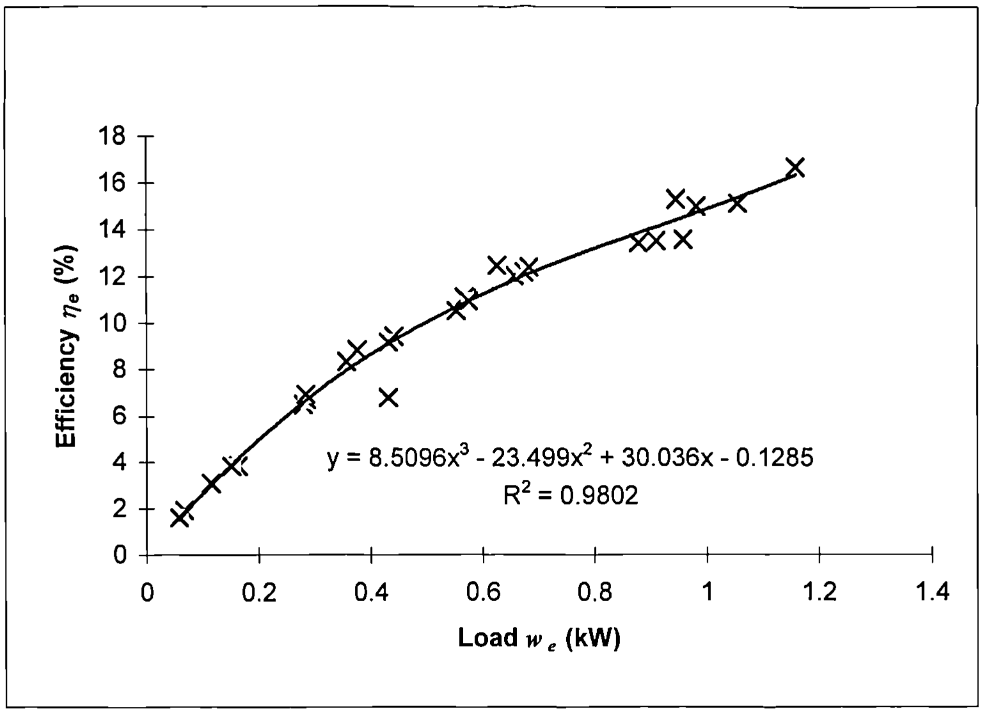


Figure 6.3 Engine/Generator Conversion Efficiency

6.2.2 EHE3 Thermal Delivery with Respect to Engine Load

Figure 6.4 describes the relationship between engine/generator output and EHE3 thermal delivery, i.e. the heat to power ratio of the CHP sub-system (λ). All the results were taken under steady state conditions.

The results obtained from the optimised plant (see Appendix E.1.1), show a favourable value for λ at low engine loads, compared to the assumed relationship employed in the *conceptual modelling* (see Section 4.4.1). The reported r^2 value for the return function validates the result.

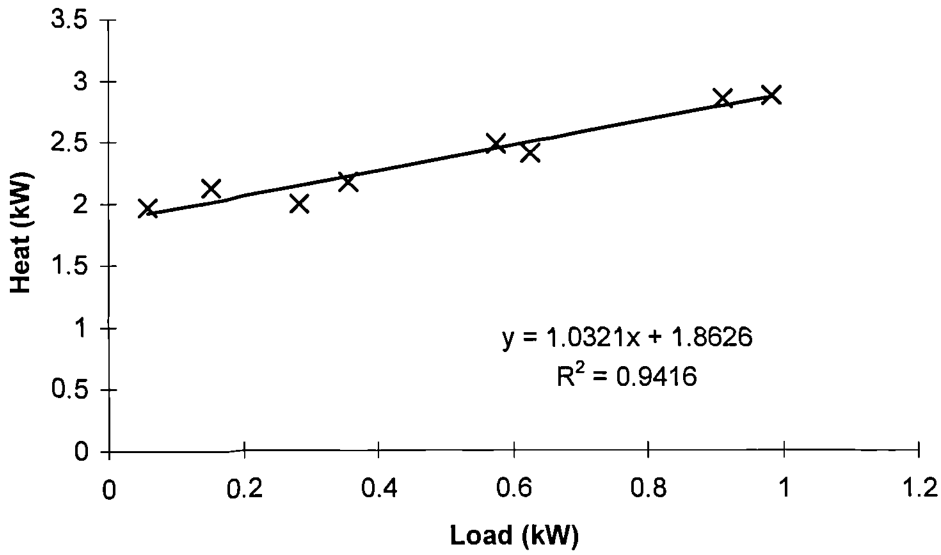


Figure 6.4 EHE3 Thermal Delivery with respect to Engine Load

6.2.3 Heat Pump Performance

Figure 6.5 shows the obtained relationship between heat pump thermal delivery and work input (i.e. heat pump COP). Although this result yields a good linear relationship, owing to thermal capacitance effects of the heat pump compressor, this result is unreliable.

Figure 6.6 compares the heat pump compressor temperature to the COP, with respect to time. As the compressor temperature increases with use, this decreases the COP correspondingly. With reference to Figure 6.65, it can be observed that lower values of COP correspond to higher work inputs. As the compressor temperature increases, it incurs greater losses – reducing COP. Hence, COP variation is primarily due to the thermal effects of the heat pump compressor. See Appendix E.1.2, for tabulated results

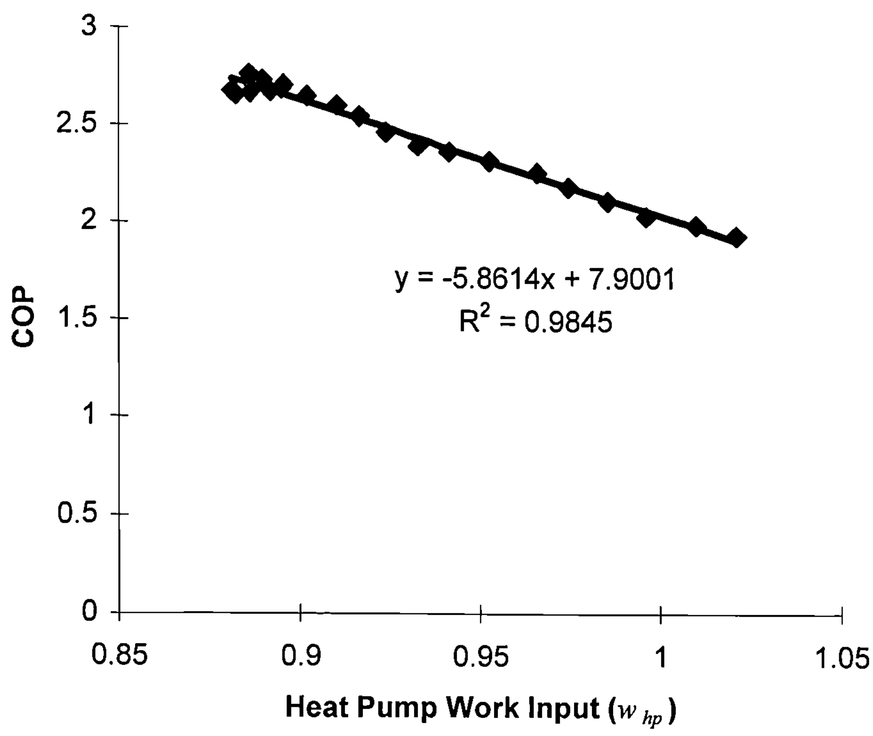


Figure 6.5 Heat Pump COP with respect to Work Input

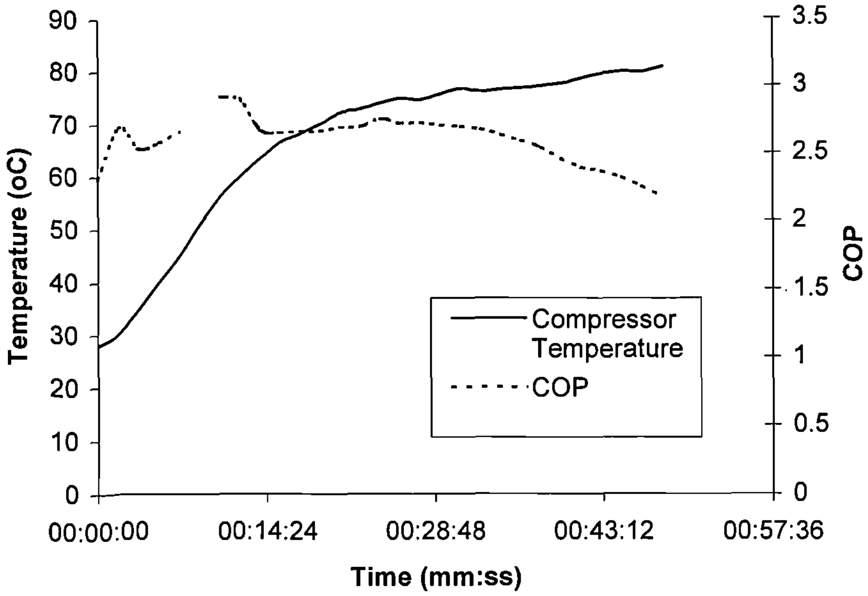


Figure 6.6 Comparison of Heat Pump Compressor Temperature and COP

6.3 First Law Analysis

This section will examine the first law performance of the prototype plant under different modes of operation. A set of representative results for each mode of operation under steady state conditions will be analysed and associated sensitivity analysis applied. All test results and development of sensitivity analysis will be found in Appendix E. The analysis will take the form of energy balances, which will quantify flows of energy through the plant.

6.3.1 CHP Mode First Law Results and Analysis

The CHP mode, described in Section 4.3.1, is essentially that of a conventional CHP plant. Figure 6.7 shows energy flow through the plant when operating in CHP mode.

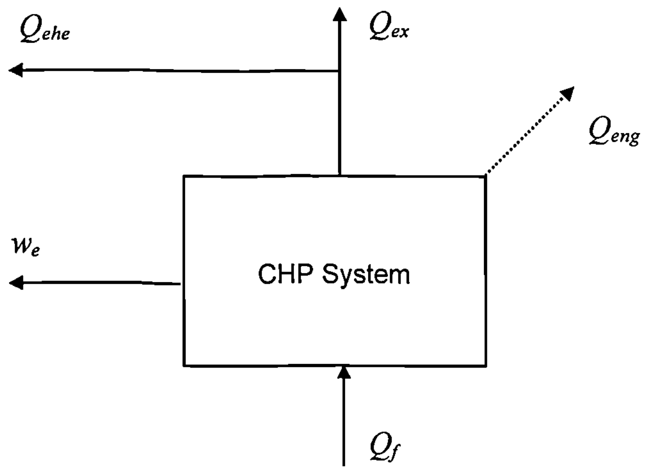


Figure 6.7 Definition of CHP Energy Balance

The energy balance gives:

$$Q_f = w_e + Q_{ehe} + Q_{eng} \qquad (6.1)$$

All the energy supplied as fuel must be recovered as useful energy (w_e and Q_{ehe}) or lost (Q_{eng}). Losses occur in the exhaust heat recovery system (Q_{ex}) and through direct radiative (Q_{rad}) and convective (Q_{conv}) heat transfer from the surface of the engine, hence:

$$Q_f = w_e + Q_{ehe} + Q_{ex} + Q_{rad} + Q_{conv} \qquad (6.2)$$

Figure 6.8 shows the steady state thermal condition of the CHP operating mode for *CHP Test 1* (from 10:57:32 to 11:00:17 – see Appendix E.1.1). The exhaust mass flow ($\dot{m}_{ex}$) is calculated in Appendix D.1.7 and will be used throughout this chapter.

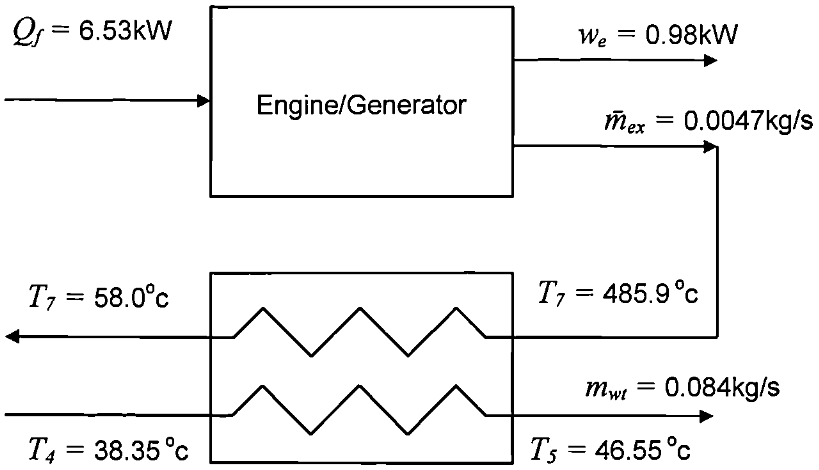


Figure 6.8 Thermal Condition of Plant in CHP Mode

The thermal delivery to the LPW system can be calculated from:

$$Q_{ehe} = \dot{m}_{wt} cp_{wt} (T_5 - T_4) \qquad (6.3)$$

Taking cp_{wt} from property tables [36]:

$$Q_{ehe} = (0.084)(4.2)(46.55 - 38.35) = 2.89kW$$

Sensitivity analysis derived in Appendix E.2.3 gives:

$$\Delta Q_{ehe} = \sqrt{\begin{matrix}[cp_{wt}(T_5 - T_4)\Delta\dot{m}_{wt}]^2 + [\dot{m}_{wt}(T_5 - T_4)cp_{wt}]^2 + \\ [\dot{m}_{wt}cp_{wt}(1 - T_4)\Delta T_5]^2 + [\dot{m}_{wt}cp_{wt}(T_5 - 1)\Delta T_4]^2\end{matrix}} \qquad (E.9)$$

Assuming that Δcp_{wt} = *0*, and applying individual transducer tolerances, ascertained in Appendix D.2, gives:

Q_{ehe} = *2.89 ±0.37 kW*

Substituting values for Q_f, Q_{ehe} and w_e in Equation 6.2 gives:

$$6.53 = 2.89 + 0.98 + Q_{ex} + Q_{rad} + Q_{conv}$$

Hence: $Q_{ex} + Q_{rad} + Q_{conv} = 2.66\ kW$

To confirm the energy balance, it is necessary to estimate the amount of available energy in the engine exhaust not recovered by the EHE. If all available exhaust energy were recovered, then the exhaust gas outlet temperature (T_7) would fall to the average water temperature inside the EHE (which is assumed to be T_5). Hence the lost energy in the exhaust gas can be calculated from:

$$Q_{ex} = \dot{m}_{ex} cp_{ex} (T_7 - T_5) \qquad (6.4)$$

Where: cp_{ex} = 0.722 [kj/kgK] - (at 500K see Appendix C.1)

$\dot{m}_{ex}$ = 0.0047 [kg/s] – (calculated in Appendix D.1.7 for full load)

$$Q_{ex} = (0.722)(0.0047)(58.04\text{-}46.55) = 0.039\ kW$$

This result indicates that the losses in the exhaust system were negligible and virtually all the energy losses occurred via convective and radiative heat transfer from the engine block. Simplified radiative heat transfer theory [37] approximates:

$$Q_{rad} = \sigma a T_{surface}^4 \qquad (6.5)$$

Where:

$\sigma = 5.669_{x10}^{-11} kW/m^2K^4$, *Stefan-Boltzmann* constant.
a = $0.15 m^2$, Surface area of engine block.
$T_{surface}$ = $473K$, Approximate average surface temperature.

Hence:

$$Q_{rad} = (5.669_{x10}^{-11}).(0.15)(473) = \mathbf{0.426kW}$$

To assess the convective heat transfer, it is necessary to use empirical relationships. It will be assumed that the engine block is approximated to a rectangular form exposed to an air flow of 25m/s, from the generator cooling fan. Calculating a film air temperature (T_{flim}), with an ambient air temperature (T_8) and the average surface temperature ($T_{surface}$) gives:

$$T_{film} = \frac{T_{surface} + T_8}{2} \approx 400K \qquad (6.6)$$

Using property table and engine data:
d = 0.1m, diameter of engine block.
C = 0.102 and n = 0.675, constants.
ρ = 0.8824 kg/m^3.
μ = 2.286$_{x10}$$^{-5}$ kg/ms.
Pr = 0.688.

From *Holman* [37]: $\mathrm{Re} = \frac{\rho.ud}{\mu}$ *(6.7)* and $\frac{hd}{k_b} = C\,\mathrm{Re}^n.\mathrm{Pr}^{\frac{1}{3}}$ *(6.8)*

Which gives:

$$\mathrm{Re} = \frac{(0.8824)(0.1)(25)}{2.286_{x10}{}^{-5}} = 0.965_{x10}{}^{5}$$

And

$$\frac{hd}{k_b} = (0.102)(0.965)^{0.675}.(0.688)^{\frac{1}{3}} = 208.46$$

Hence h = 0.07014 kW/m^{2}K

Finally, convective heat transfer can be approximated by:

$$Q_{conv} = hA(T_{Surface} - T_8) = (0.07014)(.15)(170) = 1.79kW$$

Combining convective, radiative and exhaust losses to give total losses (Q_{loss}):

$$Q_{loss} = Q_{conv} + Q_{rad} + Q_{exhaust} = 2.26\ kW \qquad (6.9)$$

This accounts for 85% of the losses. The above analysis gives only an approximation of radiative and convective losses. However, the results are of the expected order. The analysis does indicate that the majority of losses (80%) occur through convection from the engine block. This will be discussed further in Section 6.5. As the estimation of radiative losses employed simplified *black body* assumptions, the deficit of energy must be accounted for by convective losses. Hence the energy balance described by Equation 6.2 and shown in Figure 6.9 can be confirmed:

$$Q_f = w_e + Q_{ehe} + (Q_{ex} + Q_{conv} + Q_{rad}) \qquad (6.2)$$

$6.53 \approx 0.98 + 2.89 + 2.26$ (Error = 6%)

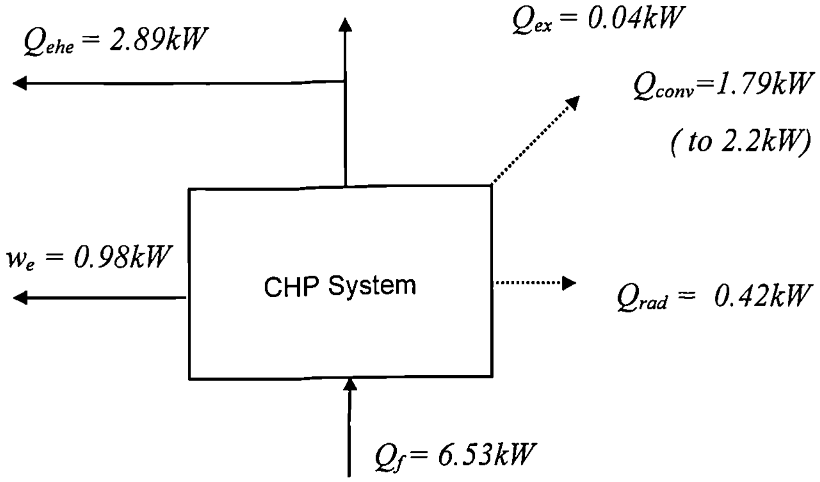


Figure 6.9 Completed CHP Experimental Energy Balance

Applying relevant theory and sensitivity analysis, the following efficiencies shown in Table 6.2 and Figure 6.10 were calculated with associated tolerances*.

Table 6.2 Results of CHP Analysis

Parameter	Function	Sensitivity Function	Value
Electrical conversion efficiency	$\eta_e = \frac{w_e}{Q_f}$	*E.6*	15% (±0.004%)
Thermal CHP efficiency	$\eta_{th} = \frac{Q_{ehe}}{Q_f}$	*E.12*	44.3% (±5.7%)
Total CHP efficiency	$\eta_{chp} = \frac{w_e + Q_{ehe}}{Q_f}$	*E.15*	59.3% (±5.7%)

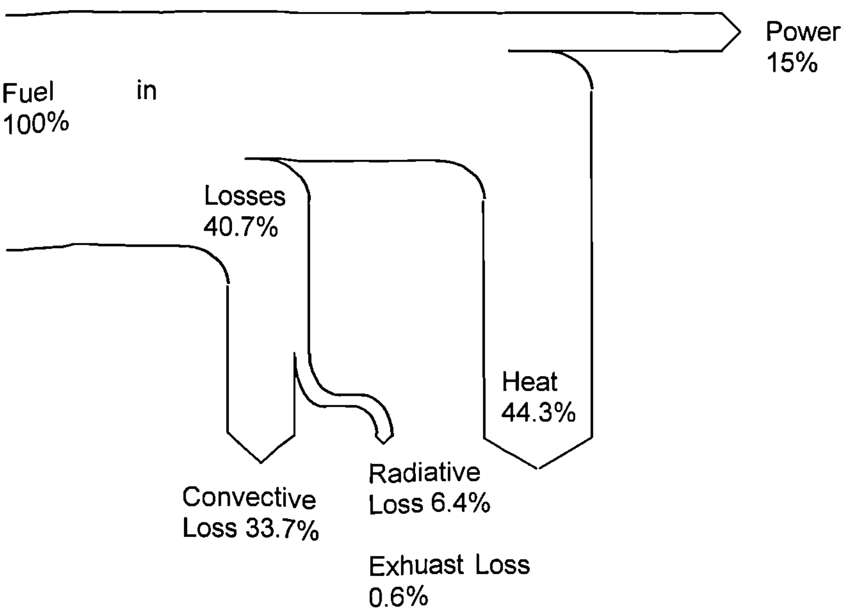


Figure 6.10 Sankey Diagram of CHP Operation

* See: Appendix E.2 for sensitivity analysis; Appendix D.2 for transducer tolerances.

6.3.2 First Law Results and Analysis of Heat Pump (HP) Mode

In the *HP* heating mode (as described in Section 4.3.2), all the generated electrical output would be utilised by the heat pump. The heat pump electrical system could not be directly connected to that of the generator, as a result of difficulties highlighted in Section 6.1.1.1. The heat pump was driven from the mains supply and generator output varied to produce an equivalent amount of power. The following analysis will consider the generator output to be driving the heat pump. This running condition simulates a gas driven heat pump.

The analysis of the *HP* mode will consider an energy balance of the complete plant. However, energy balances of individual components must be completed first. Figure 6.11 defines the energy balances and boundaries of the combined plant, which is described by Equation 6.10.

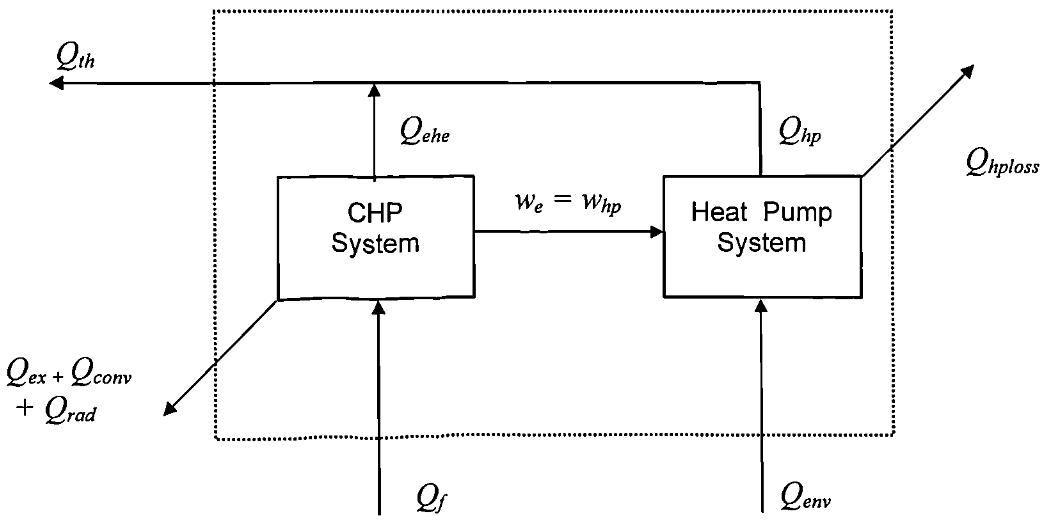


Figure 6.11 Definition of Energy Balance for *HP* mode

$$Q_f + Q_{env} = Q_{th} + (Q_{ex} + Q_{rad} + Q_{conv} + Q_{hploss}) \qquad (6.10)$$

Where the terms in brackets are considered as losses and useful delivered energy is the total thermal output (Q_{th}). The steady state thermal conditions for *HP* mode are shown in Figure 6.12, during the *HP Mode Test* (from 15:34:19 to 15:35:53 – see Appendix E.1.3).

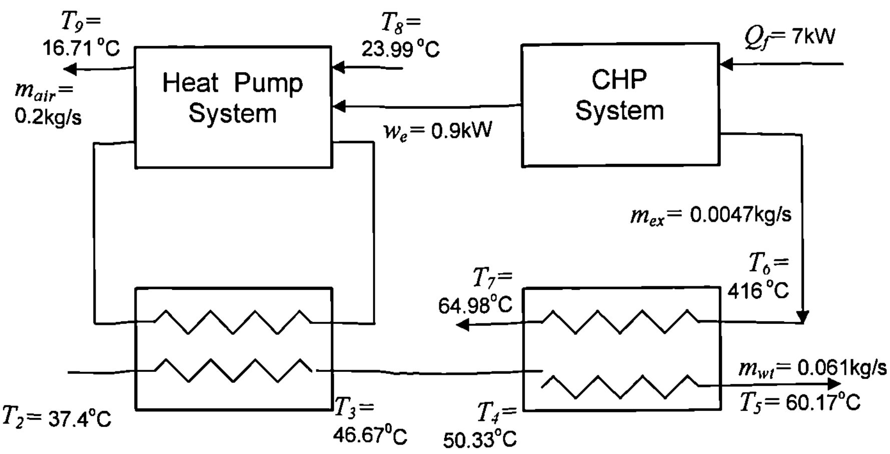


Figure 6.12 Thermal Conditions of HP Plant Operation

The energy balance (from Equation 6.2) for the CHP system gives:

$$Q_f = w_e + Q_{ehe} + Q_{ex} + Q_{conv} + Q_{rad} \tag{6.2}$$

Calculating CHP thermal output:

$$Q_{ehe} = \dot{m}_{wt} cp_{wt} (T_5 - T_4) \tag{6.3}$$

$$Q_{ehe} = (0.061)(4.2)(60.17 - 50.33) = 2.52kW$$

Applying sensivity analysis (see Appendices E.3.2. and E.3.2.7) gives:

$$Q_{ehe} = 2.52 \pm 0.45\ kW$$

Assuming that the engine losses (exhaust, radiative and convective) are approximately of the same order as those calculated in Section 6.3.1, completing the energy balance for the CHP system can be given as:

$$7 = 0.9 + 2.52 + Q_{ex} + Q_{conv} + Q_{rad}$$

Therefore $Q_{ex} + Q_{conv} + Q_{rad} = 3.58\ kW$ *(6.9)*

The higher losses experienced while running in the *HP* mode compared to those in the *CHP* mode, are due to the lower part load efficiency of the engine.

This analysis will only consider the energy balance of the heat pump system in terms of energy flow in and out of the unit and not analyse the behaviour of the refrigerant system.

The energy balance for the heat pump system is described by:

$$Q_{hp} + Q_{hploss} = Q_{env} + w_{hp} \qquad (6.11)$$

Calculating the thermal delivery of the heat pump to the LPW system from:

$$Q_{hp} = \dot{m}_{wt} cp_{wt} (T_3 - T_2) \qquad (6.12)$$

gives, with

$$Q_{hp} = (0.061)(4.2)(46.67 - 37.4) = 2.375 kW$$

Sensitivity analysis (see Appendices E.2.7 and E.3.2) gives:

$$Q_{hp} = 2.37 \pm 0.42\ kW$$

The energy acquired by the heat pump from the environmental air (Q_{env}) is calculated by considering the air flow through the air coils of the heat pump, hence:

$$Q_{env} = \dot{m}_{air} cp_{air} (T_9 - T_8) \qquad (6.13)$$

Where

$$\dot{m}_{air} = \dot{V}_{air} \rho_{air_{300K}} = (0.17)(1.177) = 0.200 kg/s \qquad (6.14)$$

$\dot{V}_{air}$ was taken from manufacturer's information (see Appendix D.1.4).
cp_{air} = 1.005 kg/kgK, from property tables.

$$Q_{env} = (0.200)(1.005)(23.99\text{-}16.47) = 1.51 kW$$

Losses from the heat pump will be primarily from the compressor, as this component is subject to mechanical friction and electrical losses. The temperature of the compressor surface (T_{hp}) at steady state running was measured at approximately at 82^{O}C (see Figure 6.6). The compressor was isolated from the main heat pump frame and so conductive losses can be considered negligible. At the recorded surface temperature, radiative heat transfer will also be negligible and the predominant form of heat transfer will be convection. To approximate thermal losses from the compressor, free convection empirical relationships will be employed. The compressor is approximated to a cylinder (with a surface area of 0.07m^2), for the following calculations.

From *Hollman* [37]: $h = 1.42\left(\frac{\Delta T}{L}\right)^{\frac{1}{4}}$ *(6.15)*

Where:

L = 0.15m, height of the compressor.

ΔT = 58, Temperature difference between convective surface (T_{hp}) and the surrounding air (T_8).

$$h = 1.42\left(\frac{58}{0.15}\right)^{\frac{1}{4}} = 6.3W/m^2K$$

Substituting into:

$$Q_{hploss} = hA(T_{hp} - T_8) \qquad (6.16)$$

Gives:

$$Q_{hploss} = (6.3_{x10}{}^{-3})(0.0638)(58) = 0.023kW$$

From Equation 6.11

$$Q_{hploss} = Q_{hp} + w_{hp} - Q_{env} = 0.035kW \qquad (6.11)$$

The losses approximated and those calculated from measured parameters are of the same order. Hence the majority of losses can be attributed to convection from the heat pump compressor. Losses lie within the error identified by the sensitivity analysis of 12.6% (see Appendix E.3.2).

Combining the thermal delivery from the *CHP* and *heat pump* system gives the total thermal output (Q_{th}) and applying the relevant sensitivity function (see Appendices E.2.11 and E.3.2):

$$Q_{th} = Q_{ehe} + Q_{hp} = 2.521 + 2.37 = 4.89 \pm 0.616kW \qquad (6.17)$$

The energy balance for the whole plant (Equation 6.10) can now be completed, as shown in Figure 6.13.

$$Q_{f+}Q_{env} = Q_{th} + (Q_{ex} + Q_{rad} + Q_{conv} + Q_{hploss}) \qquad (6.10)$$

$7 + 1.45 \approx 4.89 + 3.58$

Where Q_{hploss} is ignored.

The *coefficient of performance* (*COP*) for the heat pump is defined by:

$$COP = \frac{Q_{hp}}{w_{hp}} \qquad (6.18)$$

Which gives:

$$COP = \frac{2.37}{0.90} = 2.65$$

Applying derived sensitivity analysis (see Appendices E.2.8 and E.3.2) gives:

$COP = 2.57 \pm 0.47$

The overall plant efficiency (η_{chphp}) is not a true efficiency as it only takes into account fuel input to the plant and does not consider environmentally obtained energy. Overall plant efficiency is the effective economic efficiency, that only considers energy inputs of economic value and in this case is equivalent to a *primary energy ratio*. Hence (with application of sensitivity analysis in Appendices E.2.9 and E3.2):

$$\eta_{chphp} = \frac{Q_{th}}{Q_f} = \frac{4.89}{7} = 0.7 \pm 0.088 \qquad (6.19)$$

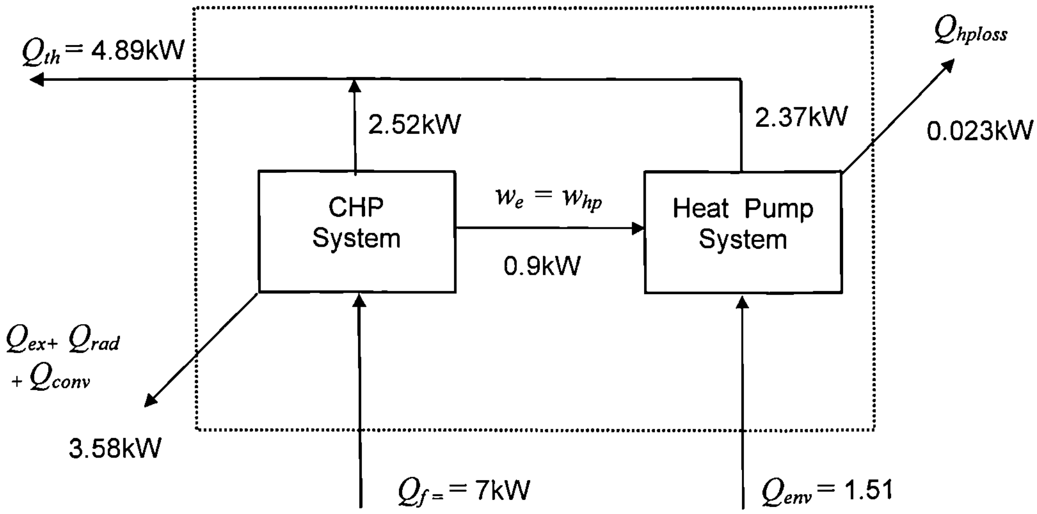


Figure 6.13 Completed Energy Balance for HP Mode

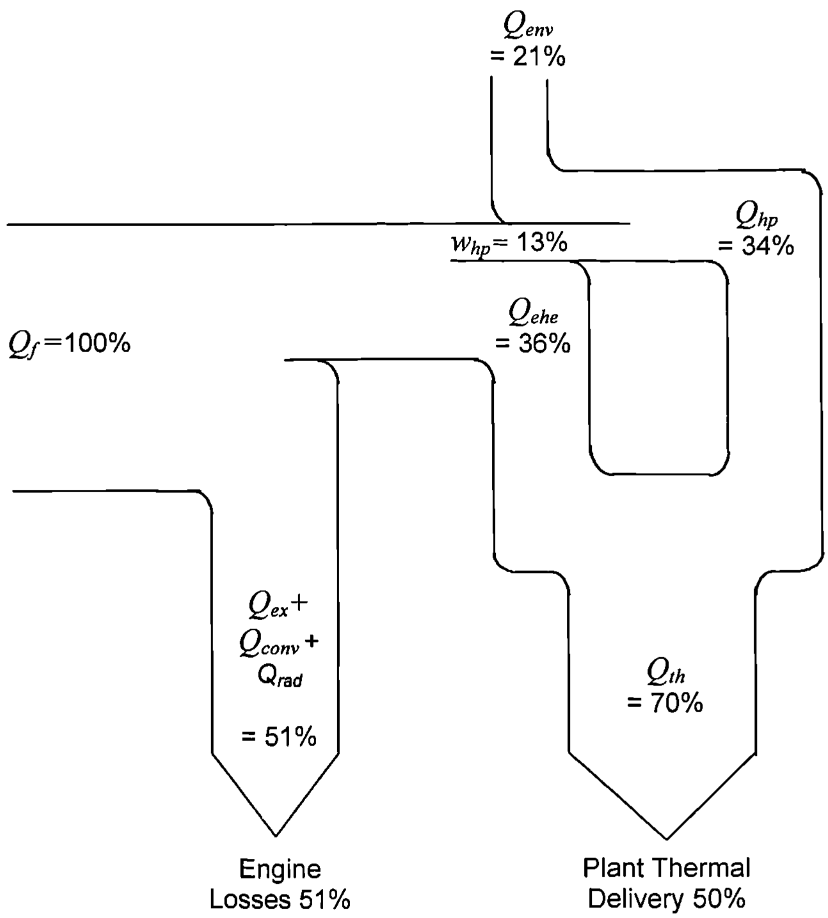


Figure 6.14 Sankey Diagram for HP Mode

6.3.3 First Law Results and Analysis of *CHP/HP* Mode

In *CHP/HP* operating mode during testing (as described in Section 4.3.3), the entire generator output had to be dissipated in resistors and the heat pump driven from mains supply (see Section 6.1.1.1). To carry out first law analysis of hybrid *CHP/HP* operation, energy balances for both CHP and HP systems will be established, before an energy balance for the entire plant is completed. Figure 6.15 and Equation 6.20 define the energy balance for the *CHP/HP* operation. The analysis of *CHP/HP* is similar to that of *HP* analysis and utilises the same sensitivity analysis.

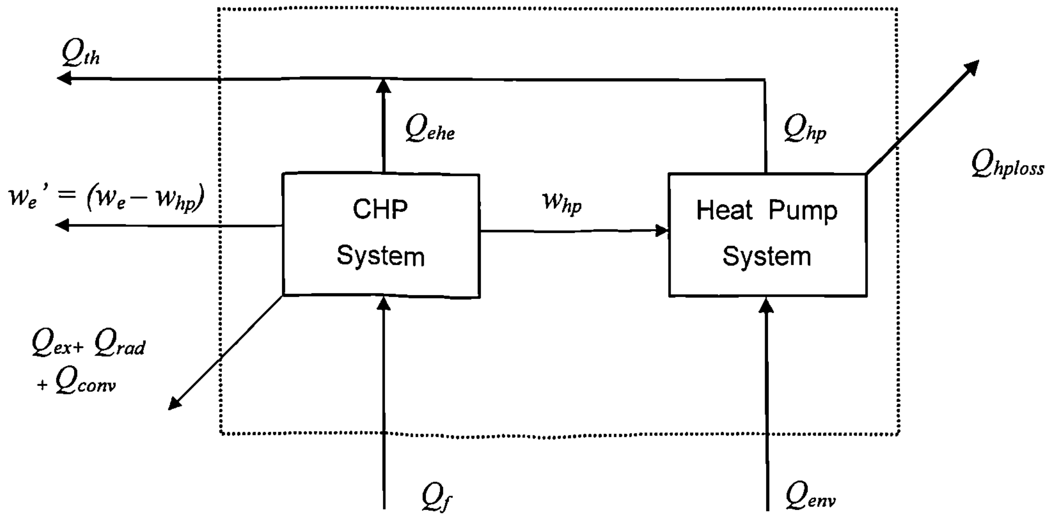


Figure 6.15 Definition of Energy Balance for CHP/HP mode

$$Q_f + Q_{env} = w_e' + Q_{th} + (Q_{ex} + Q_{conv} + Q_{rad} + Q_{hploss}) \tag{6.20}$$

Steady state results for a representative *CHP/HP* test, taken from the *CHPHP Mode Test – 1* (from 14:00:14 to 14:02:55, see Appendix E.1.4), are illustrated in Figure 6.16,

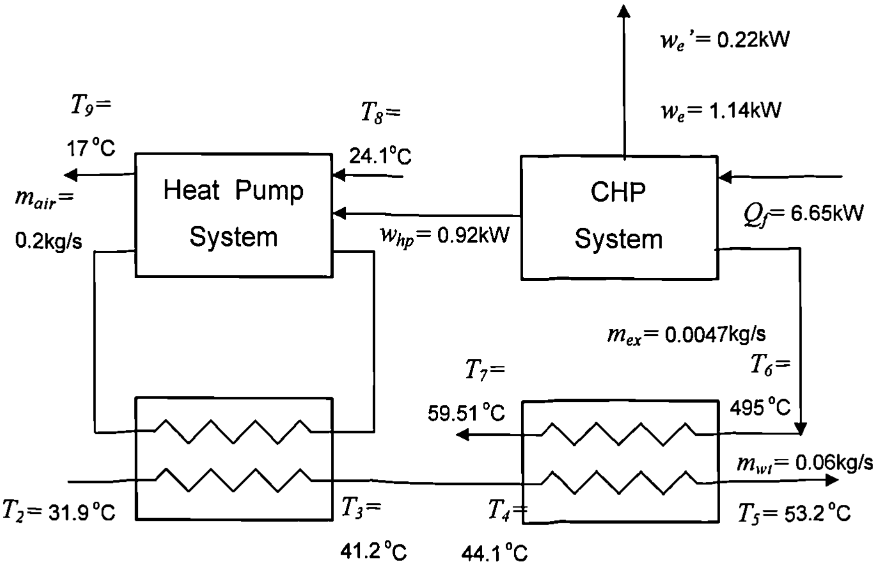


Figure 6.16 Thermal Conditions of CHP/HP Operation

From Equation 6.2, the *CHP* energy balance is described by:

$$Q_f = w_e + Q_{ehe} + (Q_{ex} + Q_{ehe} + Q_{conv}) \qquad (6.2)$$

Calculating CHP thermal output:

$$Q_{ehe} = \dot{m}_{wt} cp_{wt} (T_5 - T_4) \qquad (6.3)$$

$$Q_{ehe} = (0.06)(4.2)(53.2 - 44.1) = 2.29kW$$

Applying sensitivity analysis (see Appendices E.3.2 and E.3.3) gives:

$$Q_{ehe} = 2.29 \pm 0.41\ kW$$

Again assuming that losses from the *CHP* system are in the order of those approximated in Section 6.3.1, the *CHP* system energy balance can be completed.

$$Q_{ex} + Q_{rad} + Q_{conv} = Q_f - (w_e + Q_{ehe}) = 6.6.5 - (1.14 + 2.29) = 3.22kW$$

The equivalent electrical power to drive the heat pump (w_{hp}) can be subtracted from the generator output (w_e) to give an adjusted electrical output (w_e').

$$w_e' = w_e - w_{hp} = 1.14 - 0.92 = 0.22kW \qquad (6.21)$$

Considering the heat pump system energy balance (from Equation 6.11):

$$Q_{hp} + Q_{hploss} = Q_{env} + w_{hp} \qquad (6.11)$$

and calculating the thermal delivery of the heat pump to the LPW system from:

$$Q_{hp} = \dot{m}_{wt} cp_{wt} (T_3 - T_2) \qquad (6.12)$$

$$Q_{hp} = (0.06)(4.2)(41.2 - 31.9) = 2.34kW$$

Sensitivity analysis (see Appendices E.2.7 and E.3.3) gives:

$Q_{hp} = 2.34 \pm 0.43\ kW$.

Calculating thermal acquisition from the air (see Section 6.3.2.):

$$Q_{env} = \dot{m}_{air} cp_{air} (T_9 - T_8) \qquad (6.13)$$

$Q_{env} = (0.200)(1.005)(24.1\text{-}17.0) = 1.43kW$

By taking the losses from the heat pump compressor to be approximately the same as in Section 6.3.2, the energy balance for the heat pump system can be completed.

$$Q_{hp} + Q_{hploss} = Q_{env} + w_{hp} \qquad (6.11)$$

$$2.34 \approx 1.43 + 0.9\,2$$

Calculating the heat pump COP, with the application of sensitivity analysis (see Appendices E.2.8 and E.3.3) *COP* gives:

$$COP = \frac{Q_{hp}}{w_{hp}} = \frac{2.34}{0.92} = 2.54 \pm 0.46$$

Combining the thermal delivery from the *CHP* and heat pump systems gives:

$$Q_{th} = Q_{ehe} + Q_{hp} = 2.29 + 2.34 = 4.63 \pm 0.59\ kW \qquad (6.17)$$

See Appendices E.2.11 and E.3.3 for sensitivity analysis pertaining to this calculation.

The energy balance for the whole plant (Equation 6.20) can now be completed:

$$Q_{f+}Q_{env} = Q_{th} + w_e + (Q_{ex} + Q_{conv} + Q_{rad} + Q_{hploss}) \qquad (6.20)$$

6.65 + 1.43 = 4.63 + 0.22 (3.22 + 0.01)

The effective overall plant efficiency can now be calculated, with the application of relevant sensitivity analysis (see Appendices E.2.10 and E.3.3):

$$\eta_{chphp} = \frac{Q_{th} + w_e'}{Q_f} = \frac{4.63 + 0.22}{6.65} = 0.73 \pm 0.089 \qquad (6.21)$$

The performance of the plant in *CHP/HP* operation is summarised in Figure 6.17 and Figure 6.18.

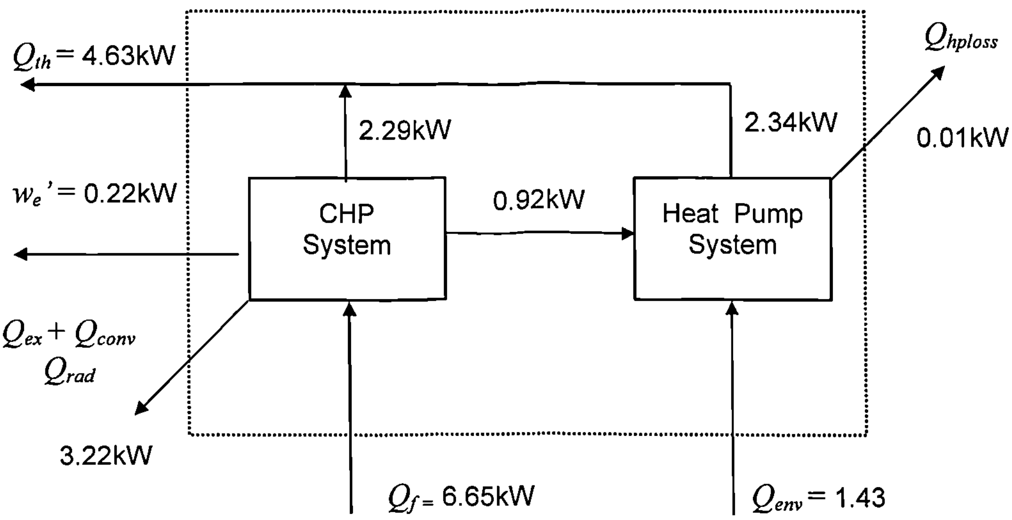


Figure 6.17 Completed Energy Balance for CHP/HP Operation

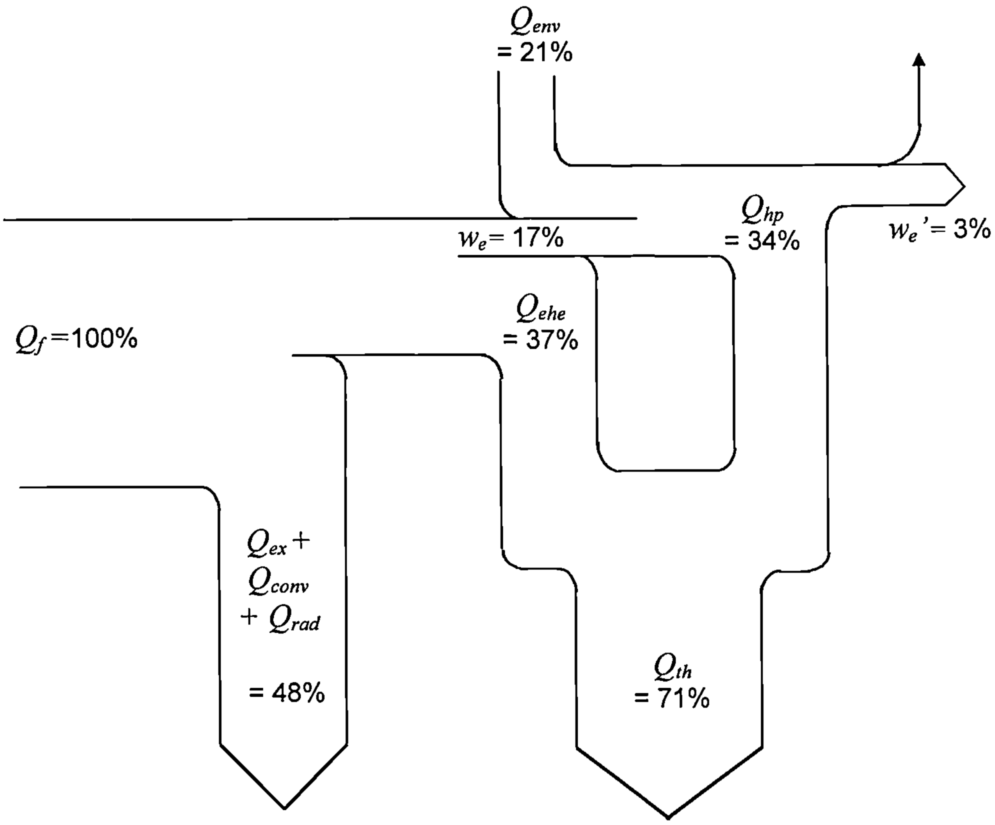


Figure 6.18 Sankey Diagram for CHP/HP Operation

6.4 Second Law Analysis of Plant Operation

Analysis of plant operation in the context of the second law of thermodynamics is contained in the following section. Second law analysis was carried out to examine aspects of plant operation that could not be dealt with using first law techniques (see Section 6.3).

By assessing the maximum available work (exergy) that could be obtained from a matter flow or energy transfer, the quality of an energy transfer can quantified. As the second law of thermodynamics implies that work energy has greater value than thermal energy, the analysis of energy quality has practical and economic implications. The exergy method of plant analysis will be employed in the following section (development of the methodology is contained in the relevant reference [38]). The exergy method examines the properties of a matter stream and calculates the maximum work that could theoretically be carried out by the matter stream by bringing it to a reference state. The exergy analysis was carried out under the following assumptions:

- Potential, kinetic, electromagnetic, and electrostatic energies are negligible and will not be included in the analysis.
- The reference state will be assumed to be the atmospheric environment at pressure (P_0) of 1 bar and at a temperature (T_0) of 298.15 K.
- The engine and heat pump will be assumed to be steady flow devices, as detailed examination of internal cycles will not contribute to the understanding of the CHP/HP concept. This will also allow control boundaries to be applied for individual components.

6.4.1 Second Law Analysis of CHP Operation

The approach taken in the second law plant analysis is to calculate the maximum work availability (hence exergy) of energy transfers and fluid streams throughout the plant. Once exergy values have been ascertained, exergy balances can be completed and second law efficiencies calculated. Figure 6.19 shows the condition of plant energy transfer and fluid condition throughout the plant in CHP operation, for *CHP Test 1* (from 10:57:32 to 11:00:17 – see Appendix E.1.1).

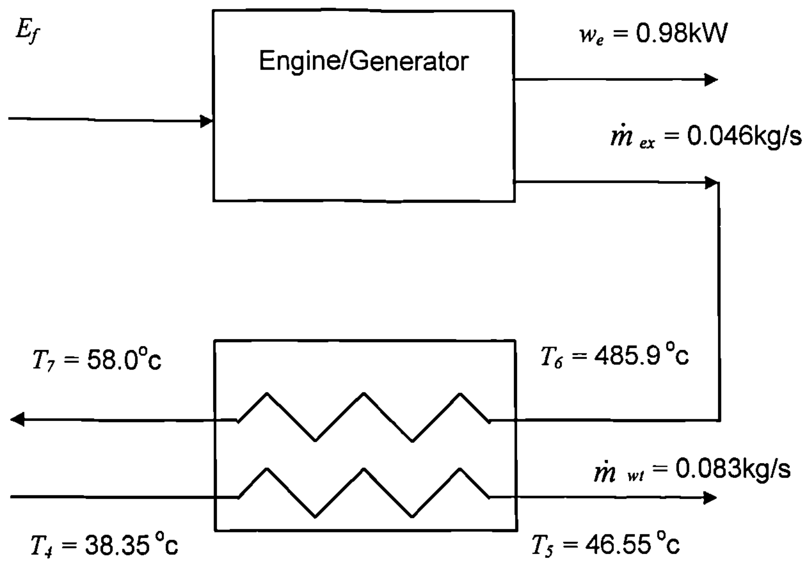


Figure 6.19 Thermal Condition of Plant under CHP Operation

Fuel Exergy Input Rate:

The exergy content of the fuel input is equivalent to the maximum work that could be carried out by ignition of the fuel under perfect reversible conditions. To simplify the analysis, the relationship identified by Szargut and Styrylska [38] will be employed, where:

$\varphi = \dfrac{\varepsilon_f}{NCV}$ (6.22), rearranging gives: $\varepsilon_f = \varphi(NCV)$ (6.23)

Multiplying both sides by the mass flow of fuel (m_f) gives:

$$\varepsilon_f = \varphi(NCV) \qquad (6.23)$$

$$E_f = \varphi Q_f \qquad (6.24)$$

Kotas [38] gives $\varphi = 1.04$

Hence the exergy fuel input (E_f) is:

$\boldsymbol{E_f = (1.04)(6.53) = 6.78\ kW}$

Exergy Content of Combustion Air

The combustion air induced into the engine is assumed to be in the same condition as the environmental reference state and hence:

$\boldsymbol{E_{air} = 0.00\ kW}$

Exergy Engine/Generator Set Electrical Output

As exergy is a measure of work availability, the exergy content of an engine/generator's electrical output will be pure exergy, hence:

$$E_e = w_e \qquad (6.25)$$

$E_e = (1.0)(0.98) = 0.98$

Exergy of LPW system at Points 4 and 5 (see Figure 6.19)

Under assumptions made in the introduction, the maximum work available in a fluid, per unit mass, can be shown to be:

$$\varepsilon_i = (h_i - h_0) - T_o(s_i - s_0) \qquad (6.26)$$

The exergy values for points 4 and 5 were calculated from:

$$E_i = \varepsilon_i \dot{m}_{wt} \qquad (6.27)$$

Entropy and enthalpy values for the LPW system at points 4 and 5 were found from property tables. Table 6.3 tabulates the calculation of E_4 and E_5.

Table 6.3 Exergy Condition of LPW System for Operation

Point	T_i	h_i	h_0	s_i	s_0	ε_i	E_i
	K	*kJ/kg*	*kJ/kg*	*kJ/kgK*	*kJ/kgK*	*kJ/kg*	*kW*
4	311.4	160.78	105.41	0.550	0.369	1.32	**0.109**
5	319.6	194.81	105.41	0.658	0.369	3.14	**0.261**

The exergy delivery rate to the LPW system is given by:

$$E_5 - E_4 = 0.261 - 0.109 = 0.152\ kW \qquad (6.28)$$

The above result can be confirmed by applying the dimensionless exergetic temperature [38] at point 5, to the first law result for Q_{th} (see Section 6.3.1). This will give the approximate exergy delivery to the LPW system:

$$E_5 - E_4 \approx Q_{chp}\left(\frac{T_5 - T_0}{T_5}\right) \qquad (6.29)$$

$$(2.89)\left(\frac{319.6 - 298.}{319.6}\right) = (2.89)(0.0676) = 0.195kW$$

As the two results have good agreement, the exergy values stated in Table 6.3 will be considered correct.

Exergy of Exhaust Gas at Points 6 and 7 (see Figure 6.19)

The exergy content of the exhaust gas is the combined exergy content of the constituent gaseous species within the exhaust mass stream, ie:

$$\varepsilon = \sum_n \varepsilon_{i_n} x_n \qquad (6.30)$$

where x_i is the mass fraction of each species. Substituting into Equation 6.30 gives:

$$E_i = \dot{m}_n \sum_0^n [h_{i_n} - h_{0_n}) - T_0 (s_{i_n} - s_{0_n})] x_n \qquad (6.31)$$

The effect of pressure on entropy values of exhaust gases is negligible and will be ignored. The calculation of the specific exergy content of the exhaust gas at points 6 and 7 is calculated in Table 6.4.

Table 6.4 Exhaust Gas Exergy Content at Points 6 and 7 for CHP Mode

	T_i	x_i	h_i	h_0	s_i	s_0	ε_1	$\dot{m}_i$	E_I
	K		*kJ/kg*	*KJ/kg*	*kJ/kgK*	*kJ/kgK*	*kJ/kg*	*Kg/s*	*kW*
$6-N_2$	758.9	*0.769*	*491.96*	*000.00*	*07.821*	*06.836*	*152.45*		
$6-H_2O$	758.9	*0.104*	*913.19*	*000.00*	*12.294*	*10.476*	*038.59*		
$6-CO_2$	758.9	*0.127*	*472.18*	*000.00*	*05.783*	*04.855*	*024.85*		
6_{total}	**758.9**	**1.000**					**215.88**	**0.0046**	**0.993**
$7-N_2$	331.0	*0.769*	*034.21*	*000.00*	*06.935*	*06.836*	*003.55*		
$7-H_2O$	331.0	*0.104*	*061.81*	*000.00*	*10.655*	*10.475*	*000.85*		
$7-CO_2$	331.0	*0.127*	*029.28*	*000.00*	*04.940*	*04.855*	*000.50*		
7_{total}	**331.0**	**1.000**					**004.90**	**0.0046**	**0.022**

Results for E_6 and E_7 can be validated with comparison to approximated values, found by assuming that the exhaust gas consists entirely of nitrogen acting in a perfect manner, allowing exergy values to be approximated by [38]:

$$\varepsilon_i \approx Cp(T_i - T_0) - T_0\left(Cp\ln\frac{T_i}{T_0} - R\ln\frac{P_i}{P_0}\right), \qquad (6.32)$$

Table 6.5 shows the comparison between entropy/enthalpy derived values and approximated values. As the entropy/enthalpy derived values (as calculated in Table 6.4) are of the same order as the approximated values, they will be assumed to be correct.

Table 6.5 Comparison of Specific Exergy Results.

Point	T_i	ε_t - (Entropy and Enthalpy Derived)	ε_t - (Approximated)
	K	*kJ/kg*	*kJ/kg*
6	758.9	215.88	194.9
7	331	4.90	1.6

The exergy transfer from the exhaust gas is represented by:

$$E_6 - E_7 = 0.993 - 0.022 = 0.971kW \tag{6.33}$$

Once exergy values have been ascertained, exergy balances for individual plant components (i.e., engine/generator set and EHE) can be completed to identify losses.

Exergy Balance for Engine/Generator Set

In general: Exergy Inputs = Exergy Output + Irreversible Losses

In this case, with reference to Figure 6.20:

$$E_f + E_{air} = E_6 + w_e + I_{engine}, \tag{6.34}$$

$6.78 + 0 = 0.99 + 0.98 + I_{engine}$

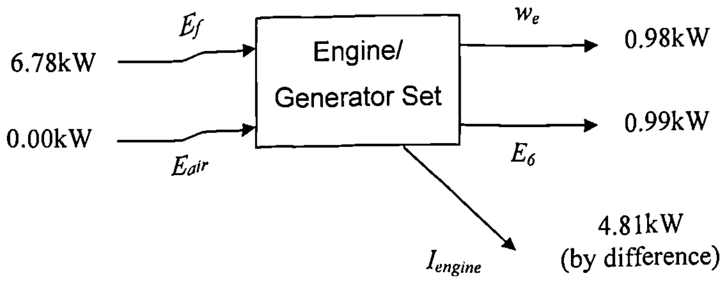


Figure 6.20 Exergy Balance for Engine/Generator in CHP Mode

Irreversible losses are those losses that could not be recovered due to irreversible processes taking place and are divided into avoidable and intrinsic losses. Intrinsic losses are due to effects such as heat transfer over a finite temperature difference or the irreversibility of real gases. This class of losses is consequential to the second law of thermodynamics and unavoidable. Avoidable losses comprise of mechanical, turbulence and electrical losses that could be practically reduced, within the individual component. Identifying irreversible losses within the engine/generator set would require extensive analysis of individual components, which would only be applicable to this particular case and would not contribute to the examination of the CHP/HP concept. At this stage, the exergy transfer via exhaust gas (E_6) will not be considered to be a loss, as it can be potentially recovered by the EHE. The exergy balance is illustrated in Figure 6.20.

Exergy Balance for EHE (see Figure 6.21)

$$E_4 + E_6 = E_5 + E_7 + I_{ehe}, \tag{6.35}$$

$$0.11 + 0.99 = 0.26 + 0.022 + I_{ehe}$$

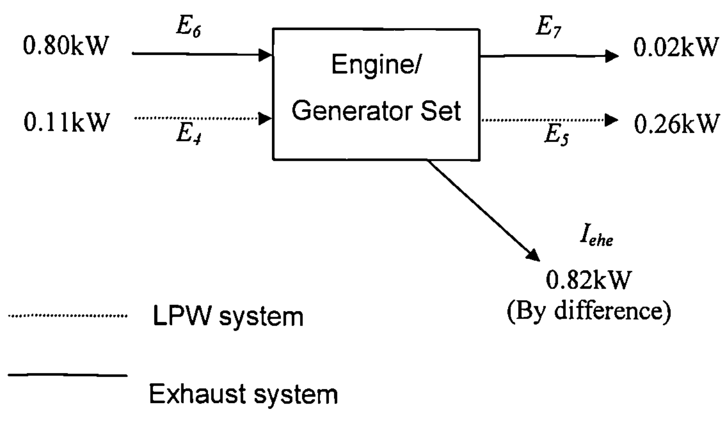


Figure 6.21 Exergy Balance for EHE in CHP Mode.

Exergy delivery to the LPW system (useful delivery) is: E_5 - E_4.

Thermal and turbulence losses from the EHE were negligible (as demonstrated in Section 6.3.1), hence irreversible losses can be considered as intrinsic. As heat transfer takes place over a wide temperature difference, the quality of the energy being transferred is significantly degraded. This degradation accounts for the intrinsic losses. This point, with its practical implications, will be discussed further in Section 6.5. As the exhaust gas leaves the system at point 7, consequently E_7 can be considered to be an avoidable loss. In order to reduce the value of E_7 (and hence the exergy loss), the LPW system would have to operate at a lower temperature. This would reduce the exergy content of the system and hence increase intrinsic losses. This point will again be discussed in Section 6.5.

The entire plant, when running in CHP mode, can now be assessed in terms of second law efficiencies. It must be noted that second law efficiencies examine the availability of an energy transfer, thus the values for second law efficiencies are lower than for their first law counterparts (see Section 6.3.1). Plant performance is summarised in Table 6.6 and Figure 6.22.

Table 6.6 Second Law Efficiencies for CHP Operation

Description	Function	Value (%)
Second law electrical efficiency.	$\psi_e = \frac{w_e}{E_f}$	14.5
Second law EHE effectiveness.	$\psi_{ehe} = \frac{E_5 - E_4}{E_6 - E_7}$	15.1
Second law CHP thermal efficiency.	$\psi_{th} = \frac{E_5 - E_4}{E_f}$	2.2
Second law total CHP efficiency.	$\psi_{thchp} = \frac{(E_5 - E_4) + w_e}{E_f}$	16.7

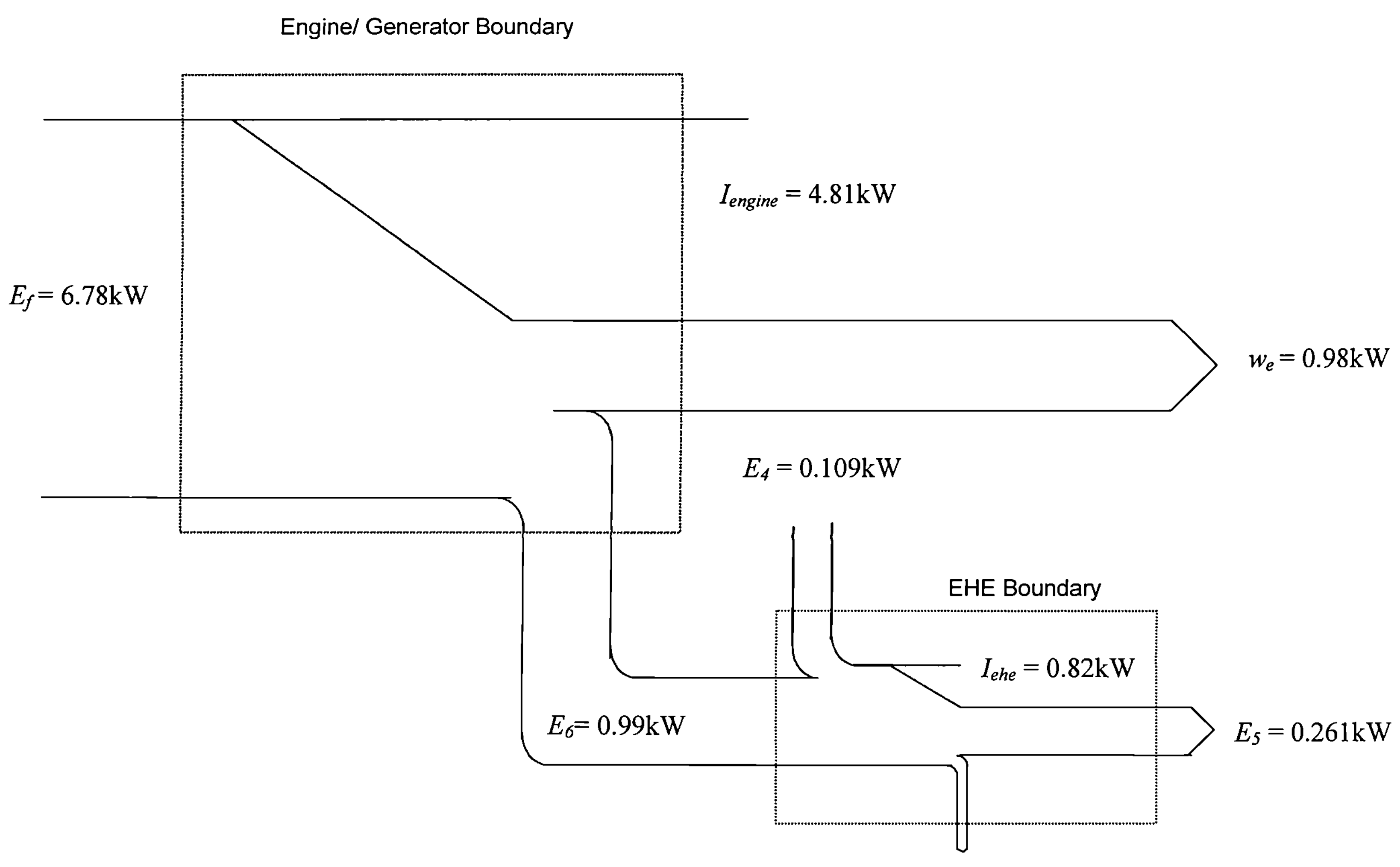


Figure 6.22 Grassman Diagram for CHP Operation

6.4.2 Second Law Analysis of Heat Pump (HP) Operation

Second law analysis of the prototype plant in HP operation was carried out in a similar manner to that of the CHP operation. Exergy values throughout the plant are first assessed and then the exergy balances are completed. The method of ascertaining exergy values from entropy and enthalpy values was validated in Section 6.4.1, and will be used in all subsequent analysis.

The thermal condition of HP operation is illustrated below in Figure 6.23 during the *HP Mode Test* (from 15:34:19 to 15:35:53 – see Appendix E.1.3). In HP operation, the entire generator electrical output is effectively consumed by the heat pump compressor: hence $w_e = w_{hp}$.

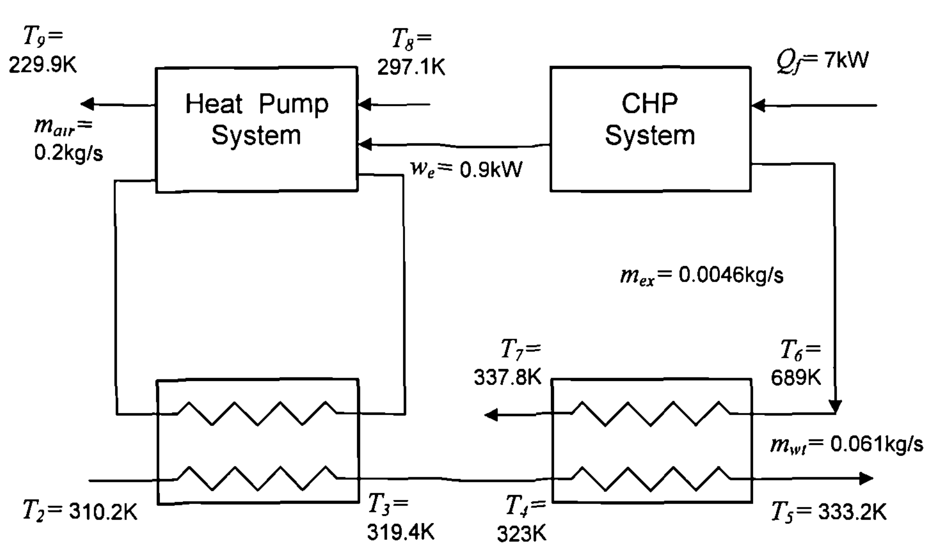


Figure 6.23 Thermal Conditions of HP Operation

Calculation of the Exergy Content of Fuel Input

As in Section 6.4.1, the Szargut and Styrylska relationship will be employed:

$$E_f = \varphi Q_f \tag{6.23}$$

E_f= (1.04)(7.00) = 7.28kW

The calculation of exergy values for fluids within the plant is tabulated in Table 6.7 and the exegetic conditions throughout the plant are illustrated in Figure 6.24.

Table 6.7 Exergy Values for HP Operation

Point	Fluid	T_i	x_i	h_I	h_0	s_I	s_0	ε_i	M_I	E_i
		K		*kJ/kg*	*KJ/kg*	*kJ/kgK*	*kJ/kgK*	*kJ/kg*	*kg/s*	*kW*
2	LPW	310.2	1.000	155.74	105.41	00.534	00.369	001.11	0.061	**0.067**
3	LPW	319.4	1.000	194.18	105.41	00.656	00.369	003.11	0.061	**0.190**
4	LPW	323.0	1.000	209.30	105.41	00.704	00.369	004.04	0.061	**0.246**
5	LPW	333.2	1.000	251.90	105.41	00.834	00.369	008.03	0.061	**0.490**
$6 - N_2$	EHE	689.0	*0.769*	*415.30*	*000.00*	*07.714*	*06.836*	*118.06*		
6 - H_2O	EHE	689.0	*0.104*	*768.50*	*000.00*	*12.095*	*10.470*	*029.54*		
$6 - CO_2$	EHE	689.0	*0.127*	*393.70*	*000.00*	*05.671*	*04.855*	*019.10*		
6	EHE	689.0	1.000					166.70	0.0046	**0.767**
$7 - N_2$	EHE	337.8	*0.769*	*041.30*	*000.00*	*06.959*	*06.836*	*003.56*		
7 - H_2O	EHE	337.8	*0.104*	*074.70*	*000.00*	*10.701*	*10.475*	*000.75*		
$7 - CO_2$	EHE	337.8	*0.127*	*035.40*	*000.00*	*04.959*	*04.855*	*000.58*		
7	EHE	337.8	1.000					004.89	0.0046	**0.022**
$8 - N_2$	HP - Air	297.1	*0.790*	*-001.09*	*000.00*	*06.832*	*06.836*	*000.18*		
$8 - O_2$	HP - Air	297.1	*0.210*	*-000.96*	*000.00*	*06.404*	*06.407*	*000.04*		
8	HP - Air	297.1	1.000					000.22	0.003	**0.001**
$9 - N_2$	HP - Air	289.9	*0.790*	*-008.57*	*000.00*	*06.804*	*06.836*	*-00.79*		
$9 - O_2$	HP - Air	289.9	*0.210*	*-007.53*	*000.00*	*06.370*	*06.407*	*-00.76*		
9	HP - Air	289.9	1.000					-01.55	0.003	**-0.005**

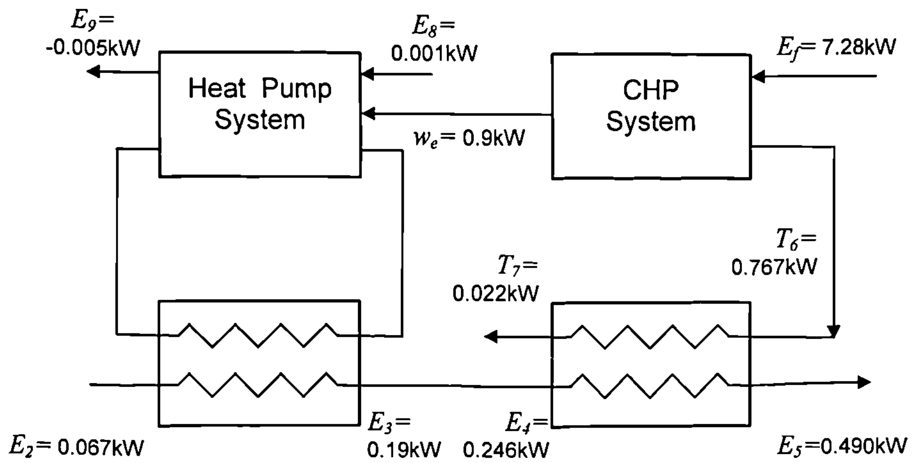


Figure 6.24 Exergetic Conditions of HP Operation

Exergy Balance for Engine Generator Set (see Figure 6.25)

$$E_f + E_{air} = w_e + E_6 + I_{engine} \tag{6.34}$$

$7.28 + 0 = 0.90 + 0.767 + I_{engine}$

By difference: $I_{engine} = 5.61\ kW$

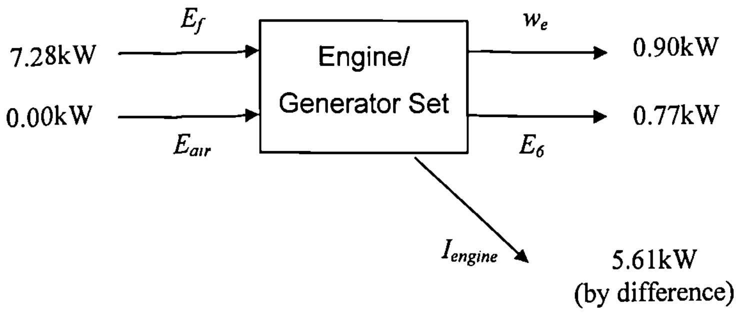


Figure 6.25 Engine/Generator Exergy Balance for HP Operation

Exergy Balance for EHE (see Figure 6.26)

$$E_4 + E_6 = E_5 + E_7 + I_{ehe} \tag{6.35}$$

$0.246 + 0.767 = 0.490 + 0.022 + I_{ehe}$

By difference: $I_{ehe} = 0.501 kW$

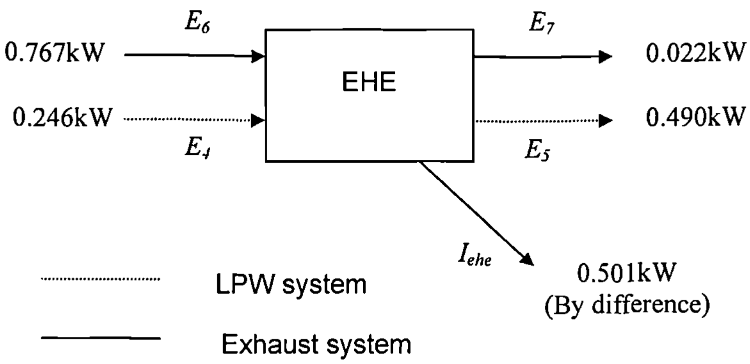


Figure 6.26 Exergy Balance for EHE in HP Operation

Exergy Balance for Heat Pump

An exergy balance for the heat pump must take account of the work input (w_{hp}) and the exergy values of fluids entering and exiting the boundary (see Figures 6.24 and 6.27), hence:

$$w_{hp} + E_2 + E_8 = E_3 + E_9 + I_{hp} \qquad (6.36)$$

Which gives: $0.9 + 0.067 + 0.001 = 0.190 - 0.005 + I_{hp}$

By difference: $I_{hp} = 0.783\ kW$.

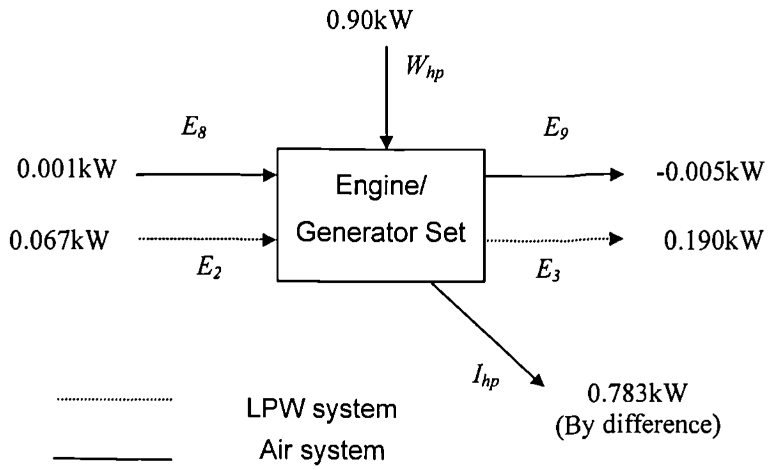


Figure 6.27 Exergy Balance of Heat Pump for HP Operation

The irreversible losses within the heat pump will be partly due to some avoidable losses such as compressor friction. However, the low exergy values associated with the LPW system imply a significant degradation of output energy quality (see Section 6.4.1). A detailed analysis of losses would involve the exergy analysis of the refrigerant system, which is beyond the scope of this analysis, as stated in the introduction to this section. It can be assumed that most of the irreversible losses experienced within the refrigerant system will be due to temperature drops across heat exchangers and expansion losses in the capillary tubes, and are therefore intrinsic in nature.

A second law coefficient of performance of the heat pump (HP) may be derived

by: $$\psi_{hp} = \frac{E_3 - E_2}{w_{hp}} \tag{6.37}$$

as the term E_3 - E_2 represents the exergy delivery from the heat pump to the LPW system. In this case:

$$\psi_{hp} = \frac{0.190 - 0.067}{0.90} = 0.137 \tag{6.38}$$

This compares poorly with the first law COP of 2.58 (see Section 6.3.2), demonstrating the degradation of energy within the system. However, by examining the equivalent exergy that could be delivered to the LPW system by direct heating, the advantages of heat pump utilisation can be demonstrated in a second law context. If the electrical energy were to be directly used to heat the LPW system, then:

$$w_{hp} = m_{wt} cp_{wt} \Delta T$$

Hence the value for T_3 that would achieved by direct heating (T_3') is:

$$T_3' = T_2 + \frac{w_{hp}}{m_{wt} cp_{wt}} \tag{6.39}$$

For the operating conditions under examination:

$$T_3' = 310.2 + \frac{0.9}{(0.06)(4.2)} = 313.77 \tag{6.40}$$

By examining the entropy and enthalpy values for water at this temperature, an effective base exergy value that could be achieved by direct heating (E_3') can be found, where $E_3' = 0.101 kW$. Hence the maximum exergy delivery that could be achieved by direct heating is:

$E_3'-E_2 = 0.101 - 0.067 = 0.034 kW$

Comparing the above value with that due to heat pump exergy delivery ($0.123 kW$), shows a 276% increase, thus demonstrating the effectiveness of the heat pump in second law terms.

Table 6.8 below summarises overall plant performance in second law terms. Although in the HP mode no electrical output leaves the plant, the second law electrical conversion efficiency for the engine/generator set has been included for reference with other modes of operation. The second law thermal efficiency of the plant considers the combined heat pump and exhaust exergy delivery to the LPW system. In the HP mode, this is identical to the total plant second law efficiency.

Figure 6.28 illustrates the exergy flows through the plant. The actual exergy values for E_8 and E_9 are negligible, owing to the air temperature being approximately the same as the reference temperature, but are represented on Figure 6.28 to illustrate the exergy flow. Due to transducer inconsistencies, a discrepancy exists between E_3 and E_4, as these values should be equal. This is accounted for in Figure 6.28, by stating their individual values at the appropriate control boundary. The transducer inconsistencies result in a 0.4% error in the overall plant exergy balance, shown in Figure 6.28.

Table 6.8 Second Law Efficiencies for HP Operation

Description	Function	Value (%)
Second Law electrical efficiency.	$\psi_e = \dfrac{w_e}{E_f}$	12.4
Second Law EHE effectiveness.	$\psi_{ehe} = \dfrac{E_5 - E_4}{E_6 - E_7}$	32.8
Second Law Heat Pump thermal efficiency.	$\psi_{hp} = \dfrac{E_3 - E_2}{w_{hp}}$	13.7
Second Law plant thermal efficiency.	$\psi_{th} = \dfrac{(E_3 - E_2) + (E_5 - E_4)}{E_f}$	5.04
Second Law total plant efficiency.	$\psi_{chphp} = \dfrac{(E_3 - E_2) + (E_5 - E_4)}{E_f}$	5.04

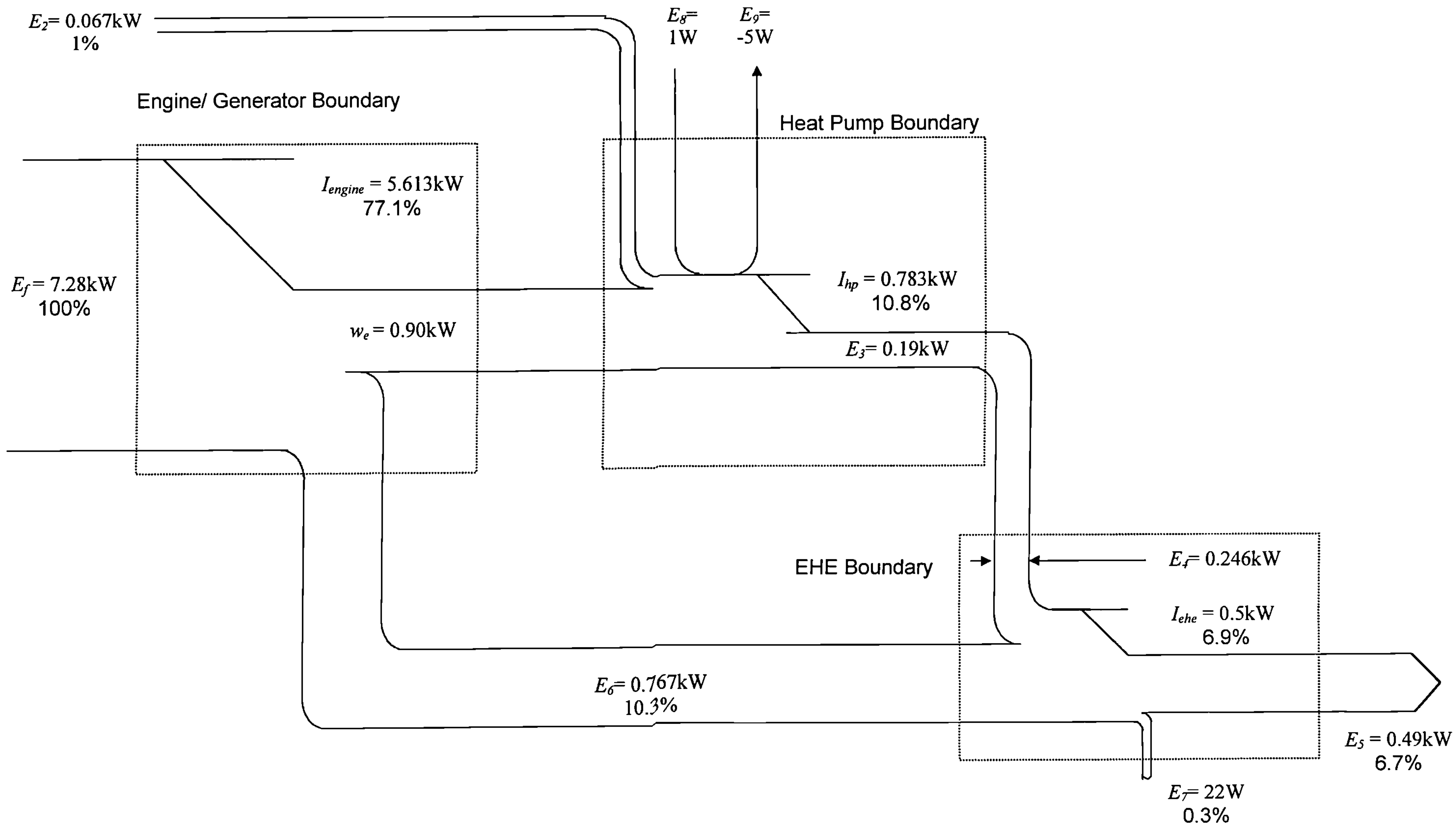


Figure 6.28 Grassman Diagram for HP Operation

6.4.3 Second Law Analysis of CHP/HP Operation

The analysis of the CHP/HP operation is similar to that for HP mode, with the exception that some electrical generation is exported from the plant (w_e') and the remainder is effectively used by the heat pump (w_{hp}), i.e.:

$$w_e = w_{hp} + w_e' \qquad (6.41)$$

The thermal condition of the plant in CHP/HP operation during the *CHPHP Mode Test – 1* (from 14:00:14 to 14:02:55, see Appendix E.1.4) is shown below in Figure 6.29. The calculation of exergy values for fluid flows throughout the plant is summarised in Table 6.9 and illustrated in Figure 6.30. Subsequent analysis will take the form of exergy balances to estimate irreversible plant losses.

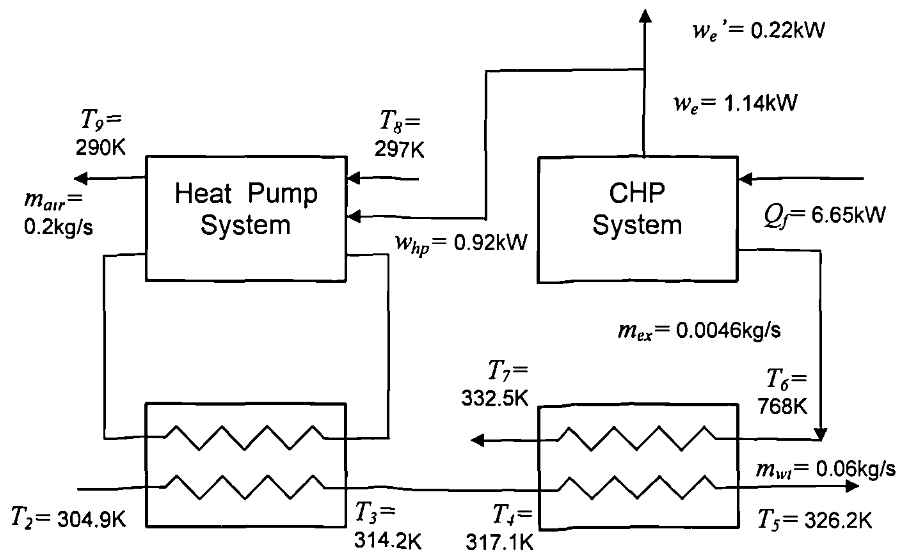


Figure 6.29 Thermal Conditions of CHP/HP Operation

Exergy Value of Fuel

Using:

$$E_f = \varphi Q_f \qquad (6.23)$$

E_f= (1.04)(6.65) = 6.916kW

Table 6.9 Exergy Values for Plant Fluids in CHP/HP Operation

Point	Fluid	T_i	x_i	h_i	h_0	s_i	s_0	ε_i	m_i	E_i
		K		kJ/kg	kJ/kg	kJ/kgK	kJ/kgK	kJ/kg	kg/s	kW
2	LPW	304.9	1.000	133.60	105.41	00.463	00.369	000.31	0.06	**0.019**
3	LPW	314.2	1.000	172.50	105.41	00.588	00.369	001.77	0.06	**0.106**
4	LPW	317.1	1.000	184.60	105.41	00.626	00.369	002.51	0.06	**0.150**
5	LPW	326.2	1.000	222.70	105.41	00.745	00.369	005.22	0.06	**0.313**
6 - N_2	EHE	768.0	*0.769*	*501.95*	*000.00*	*07.835*	*06.836*	*156.84*		
6 - H_2O	EHE	768.0	*0.104*	*932.12*	*000.00*	*12.321*	*10.476*	*039.71*		
6 - CO_2	EHE	768.0	*0.127*	*462.08*	*000.00*	*07.322*	*06.407*	*024.04*		
6	EHE	768.0	1.000					220.59	0.0046	**1.015**
7 - N_2	EHE	332.5	*0.769*	*035.77*	*000.00*	*06.940*	*06.836*	*003.72*		
7 - H_2O	EHE	332.5	*0.104*	*064.64*	*000.00*	*10.663*	*10.476*	*000.91*		
7 - CO_2	EHE	332.5	*0.127*	*030.63*	*000.00*	*04.944*	*04.855*	*000.52*		
7	EHE	332.5	1.000					005.15	0.0046	**0.024**
8 - N_2	HP - Air	297.0	*0.790*	*-001.20*	*000.00*	*06.831*	*06.836*	*000.20*		
8 - O_2	HP - Air	297.0	*0.210*	*-001.05*	*000.00*	*06.403*	*06.407*	*000.04*		
8	HP - Air	297.0	1.000					000.24	0.002	**5×10^{-4}**
- N_2	HP - Air	290.0	*0.790*	*-008.47*	*000.00*	*06.802*	*06.836*	*001.42*		
9 - O_2	HP - Air	290.0	*0.210*	*-007.44*	*000.00*	*06.377*	*06.407*	*000.33*		
9	HP - Air	290.0	1.000					001.75	0.002	**-3.5×10^{-3}**

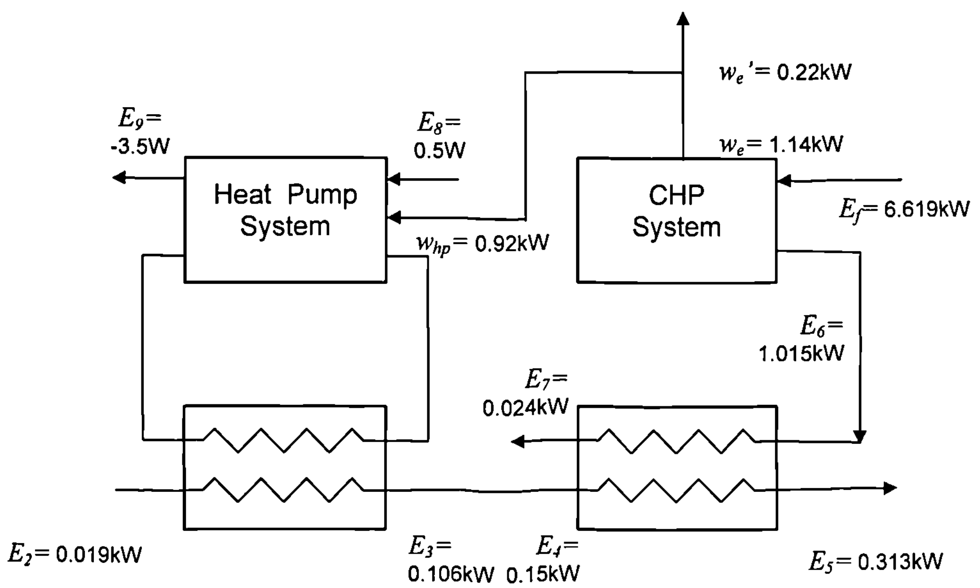


Figure 6.30 Exergetic Condition of Plant in CHP/HP Operation

Exergy Balance for Engine Generator Set (see Figure 6.31)

$$E_f + E_{air} = w_e + E_6 + I_{engine} \tag{6.34}$$

$6.916 + 0 = 1.14 + 1.015 + I_{engine}$

By difference: $I_{engine} = 4.761\ kW$

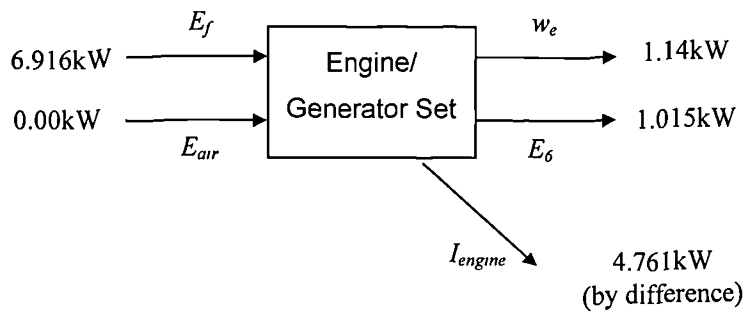


Figure 6.31 Engine/Generator Exergy Balance for CHP/HP Mode

Exergy Balance for EHE (see Figure 6.32)

$$E_4 + E_6 = E_5 + E_7 + I_{ehe} \tag{6.35}$$

$0.15 + 1.015 = 0.313 + 0.024 + I_{ehe}$

By difference: $I_{ehe} = 0.828kW$

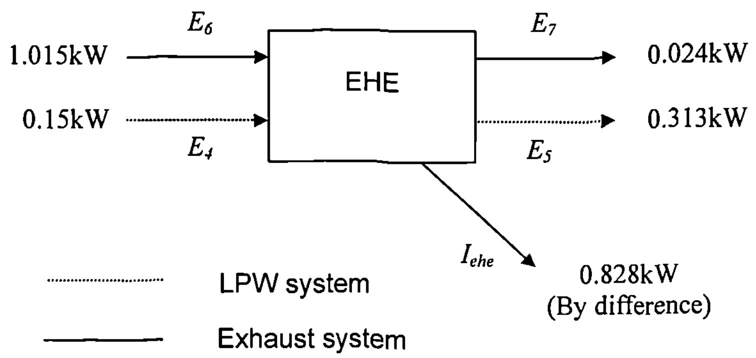


Figure 6.32 EHE Exergy Balance for CHP/HP Mode

Exergy Balance for Heat Pump (see Figure 6.33)

$$w_{hp} + E_2 + E_8 = E_3 + E_9 + I_{hp} \tag{6.36}$$

Which gives:

$0.92 + 0.019 + 0.0005 = 0.106 - 0.0035 + I_{hp}$

By difference: $I_{hp} = 0.837\ kW$.

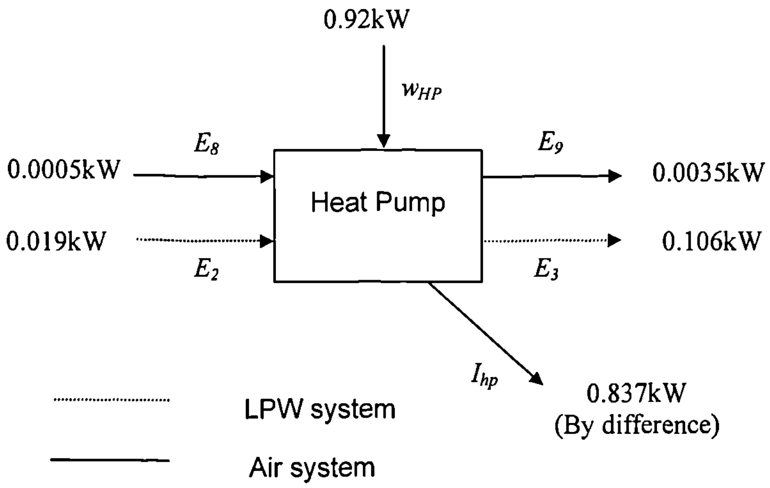


Figure 6.33 Heat Pump Exergy Balance for CHP/HP Operation

The second law effectiveness of the heat pump can be compared again to that of direct electrical heating. The LPW system temperature that could be achieved by direct heating (T_3') was calculated in the manner shown in Section 6.4.2. Table 6.10 compares the effectiveness of the exergy delivery of the heat pump to that of direct heating. The exergy delivery to the LPW system is found from: E_3-E_2.

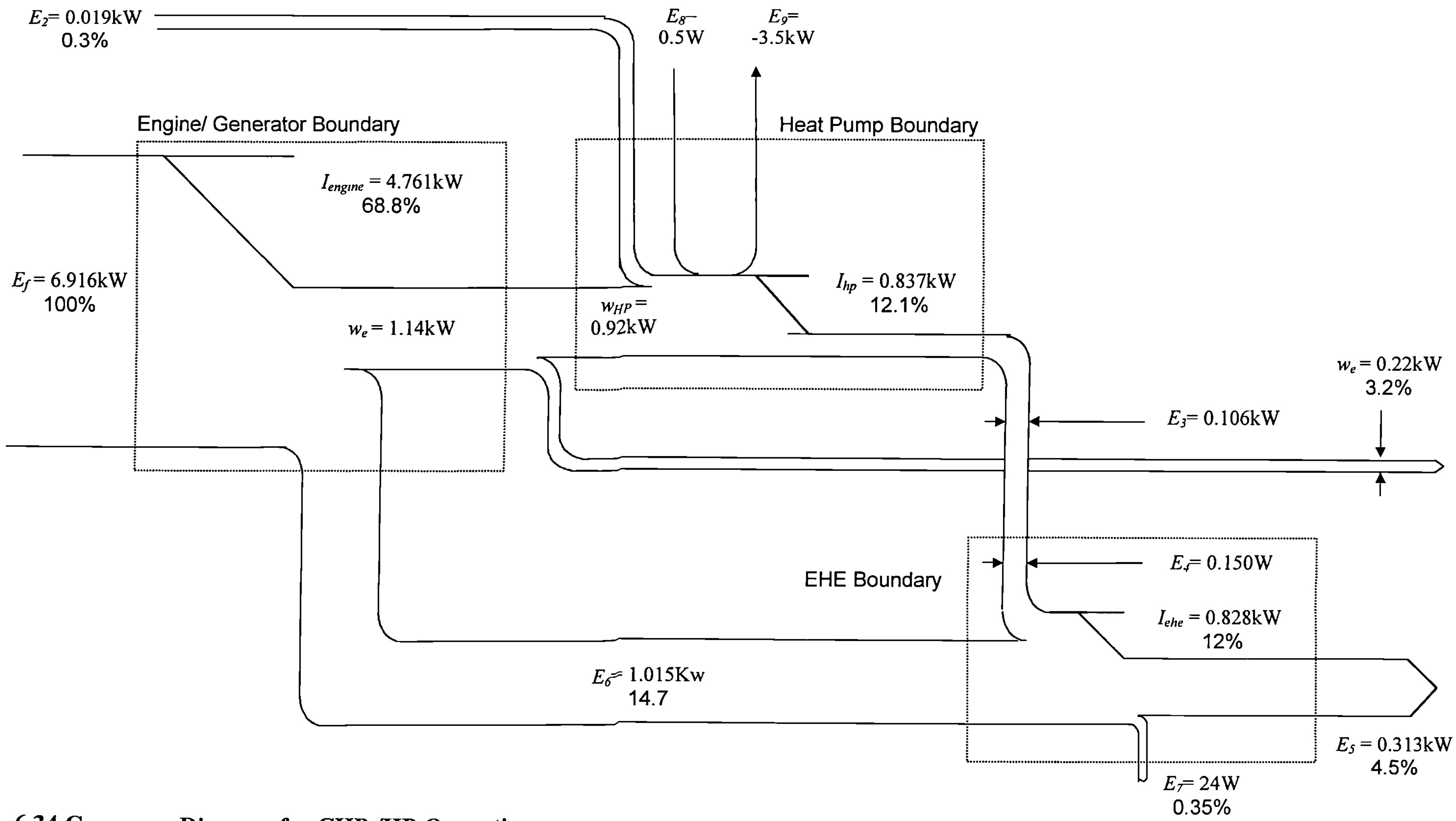
Table 6.10 Comparison of Heat Pump and Direct Heating Exergy Delivery

	T	ε	E	Exergy Delivery
	K	kJ/kg	kW	kW
Heat Pump	314.2	1.77	0.11	0.09
Direct Heating	308.6	0.83	0.05	0.03

Second law efficiencies for individual sub-systems and overall plant performance are summarised in Table 6.11. The overall second law efficiency for the plant includes the effective plant electrical output ($w_e' = w_e - w_{hp}$). The exergetic transfers throughout the plant are illustrated in Figure 6.34. As in Section 6.4.2, inconsistencies between E_3 and E_4 are accounted for.

Table 6.11 Second Law Efficiencies for CHP/HP Operation

Description	Function	Value (%)
Second Law electrical efficiency.	$\psi_e = \dfrac{w_e}{E_f}$	16.5
Second Law EHE effectiveness.	$\psi_{ehe} = \dfrac{E_5 - E_4}{E_6 - E_7}$	16.5
Second Law Heat Pump thermal efficiency.	$\psi_{hp} = \dfrac{E_3 - E_2}{w_{hp}}$	9.5
Second Law plant thermal efficiency.	$\psi_{th} = \dfrac{(E_3 - E_2) + (E_5 - E_4)}{E_f}$	3.6
Second Law total plant efficiency.	$\psi_{chphp} = \dfrac{(E_3 - E_2) + (E_5 - E_4) + w_e'}{E_f}$	6.8



6.34 Grassman Diagram for CHP /HP Operation

6.5 Discussion of Results

The following section discusses the analysis of the prototype plant in relation to practical issues. It must be noted that second law efficiencies report a lower figure than first law results. This is a consequence of exergy being a function of available work and the practical restrictions placed on the plant.

6.5.1 Identification of Losses

The use of exergy analysis allows for the identification of losses that degrade the quality of energy transfer. Table 6.12 summarises the irreversible losses for each analysed component, for each mode of operation, in terms of a percentage of the total fuel exergy input.

Table 6.12 Irreversible Losses of Plant Components

	CHP	HP	CHP/HP
	%	%	%
Engine /Generator	73.7	79.1	71.1
EHE	9.3	4.9	9.2
Heat Pump	-	10.6	12

The greatest losses are experienced in the engine/generator set and are comprised of intrinsic and avoidable losses, as stated in Section 6.4.1. Analysis of the losses would require further work. First law analysis (see Section 6.3) has previously shown that the engine/generator set has poor performance: this point is reiterated by second law analysis. Little could be done to increase the performance of the engine used, hence reduction of irreversible losses would require replacement of the engine/generator set. Water cooling the engine would result in a small reduction of exergy loss, as the exergy recovery would take place at a relatively low temperature. First law analysis does not take into account the quality of energy transfer and therefore would show a large reduction in losses with water cooling.

The Versatemp heat pump has good first law performance, with COPs in the region of 2.5 - 3. This represents the best performance that could be practically obtained from a production unit. First law analysis has indicated that compressor losses are small and so it must be concluded that irreversible heat pump losses (I_{hp}) are largely intrinsic in nature (see Section 6.4.2). The electrical energy consumed by the heat pump (considered to be pure work) is degraded to a larger amount of low quality thermal energy, giving rise to the intrinsic losses. The reduction of heat pump exergy would require a higher LPW system temperature, but as this is limited to 43°C by the refrigerant system, a new heat pump would be required. As discussed in Section 6.4.1, the irreversible losses from the EHE are intrinsic.

6.5.2 Effect of Temperature

Exergy values are dependant on temperature and as the temperatures within the LPW system are close to the reference value, the exergy values are highly sensitive to relatively small changes in temperature.

By comparing EHE performance for the CHP and HP modes, the effects of temperature are apparent. The lowest EHE irreversible losses are recorded for HP operation (see Figure 6.26), where LPW system temperatures are relatively high and the exhaust temperature is relatively low. Heat transfer takes place over a relatively narrow temperature range, which reduces the degradation of thermal energy (giving rise to lower irreversible losses). The heat transfer takes place over a wider temperature range in CHP operation, with relatively high exhaust temperatures and low LPW system temperatures. This wider range degrades the transferred thermal energy to a greater extent and produces higher irreversible losses than for the HP mode.

To reduce exergy losses, the temperature range across the heat exchanger must be reduced, by either increasing LPW system temperatures or by reducing exhaust temperatures. However, narrowing the temperature range will decrease first law performance.

In the HP operation, the lower exhaust temperature was due to the lower engine load. This was accompanied by an increased exergy loss within the engine - the highest engine/generator losses were recorded during HP operation. The overall effect is a decrease in plant performance, as the increased exergy delivery to the LPW system is negated by increased engine exergy losses.

6.5.3 Practical Implications of Exergy Analysis

In order to reduce thermal exergy losses, the LPW system temperature must rise. However, this is practically limited by the constraints of heating systems - a domestic heating system operates between 50 to 100°C. Also the application of exergy analysis to a heating system can be misleading, as the desired product is not work but heat. Application of exergy analysis to domestic CHP systems should be limited to identifying avoidable losses before the delivery of thermal energy to a dwelling.

7. Concept Evaluation Model

The following chapter will detail the development, structure and validation of a thermally based evaluation model for the domestic scale CHP/HP concept.

7.1 Introduction

For both domestic scale CHP and CHP/HP to be fully evaluated, it was necessary to construct an evaluation model that accurately simulated the prototype plant. With such a model, the operation of the prototype plant can be predicted for extended periods, which are not obtainable in the laboratory (see Section 6.1.1.2.). Additionally, numerous running conditions can be simulated, which could not be carried out practically, owing to time constraints, and parameters can be easily modified without extensive physical modifications. Data from the prototype testing was used to construct an evaluation model and later used for the validation of that model.

The evaluation model differs considerably from the earlier preliminary model. The preliminary model only considers the actual energy demand and plant output, while the evaluation model considers thermal conditions of the modelled plant - allowing for thermal capacitance to be taken into account. The preliminary model relied on many assumptions and manufacturer's claims, while the evaluation model uses experimentally derived results to simulate plant operation.

The aims of the model are:

- To simulate the extended running of the prototype plant over 24 hour periods.
- To assess the prototype CHP/HP plant in financial and environmental terms.
- To indicate areas of greatest potential improvement and to assess potential improvements.
- To test the sensitivity of financial/environmental performance with respect to varying parameters (e.g. fuel price) and component characteristics.

7.2 Model Overview

The following section will provide an overview, prior to a detailed explanation, of the evaluation model and its development.

Each component of the prototype plant is represented within the evaluation model as a sub-routine and is called in such a way to as mimic the actual prototype plant operation. The main program loop driving the evaluation model follows the flow of a unit of heating water (within the LPW system) through individual components. As a unit mass of heating water enters a component, the relevant sub-routine is called and performs the calculation pertaining to that item. This method calculates thermal plant output by considering the temperature and flow rate of the heating water. Other parameters, such as fuel consumption, are calculated from component characteristics, represented by additional sub-routines derived from experimental results. These sub-routines are called by the main program loop when data is required. The time step size of the model is dictated by the main program loop.

Electrical and thermal demand data is imported from an external database. This method allows for different demand profiles to be used without modifying the program structure. Demand data is supplied to appropriate sub-routines which then respond by matching the demands. The flow of demand data supplied to the model is controlled by the main program loop.

The economic and environmental performance of the simulated plant is monitored at each time step. If the simulated plant should not be running economically, it is shut down until conditions allow for cost-effective operation. Plant and economic parameters can be easily altered to simulate plant operation under differing conditions.

7.3 Model Structure

The following sections will detail the structure of the model by describing individual sub-routines. As the main program loop (designated *run_loop*) calls all other sub-routines, it will be discussed first.

As described previously, the *run_loop* sub-routine has three main functions:

- To load consumption data from the external database.
- To call sub-routines to calculate component response to the demand requirements.
- To analyse and present results.

The main program loop (*run_loop*) consists of two loops, a primary data control which regulates the flow of demand data into the model and a secondary loop dictated by the time step size in which calculation and analysis are carried out (see Figure 7.1). This is necessary owing to the difference in the relative sizes of demand data resolution (of the order of 30 minutes) and the model time step size (of the order of 5 seconds). This technique also allows for data resolution to be altered without significant modification of the model. See Appendix F.2 for code listings.

On starting, the model loads the demand data from the requested data set, and prepares the model to receive data to the resolution (R) and duration (n_l) of that data set. This is carried out in the *data_header* sub-routine (see Appendix F.3).

Subsequent to model preparation, the secondary loop initiates and runs at the required time step size (τ) until the maximum number of iterations has been carried out for one data period (i.e. the resolution of the demand data set - R). Once the secondary loop has been completed, then the demand data for the following data period is loaded and the secondary loop is re-initiated. Within the secondary loop, the analytical sub-routines are called. At the termination of the secondary loop, the analysis for the previous data time period is recorded and presented.

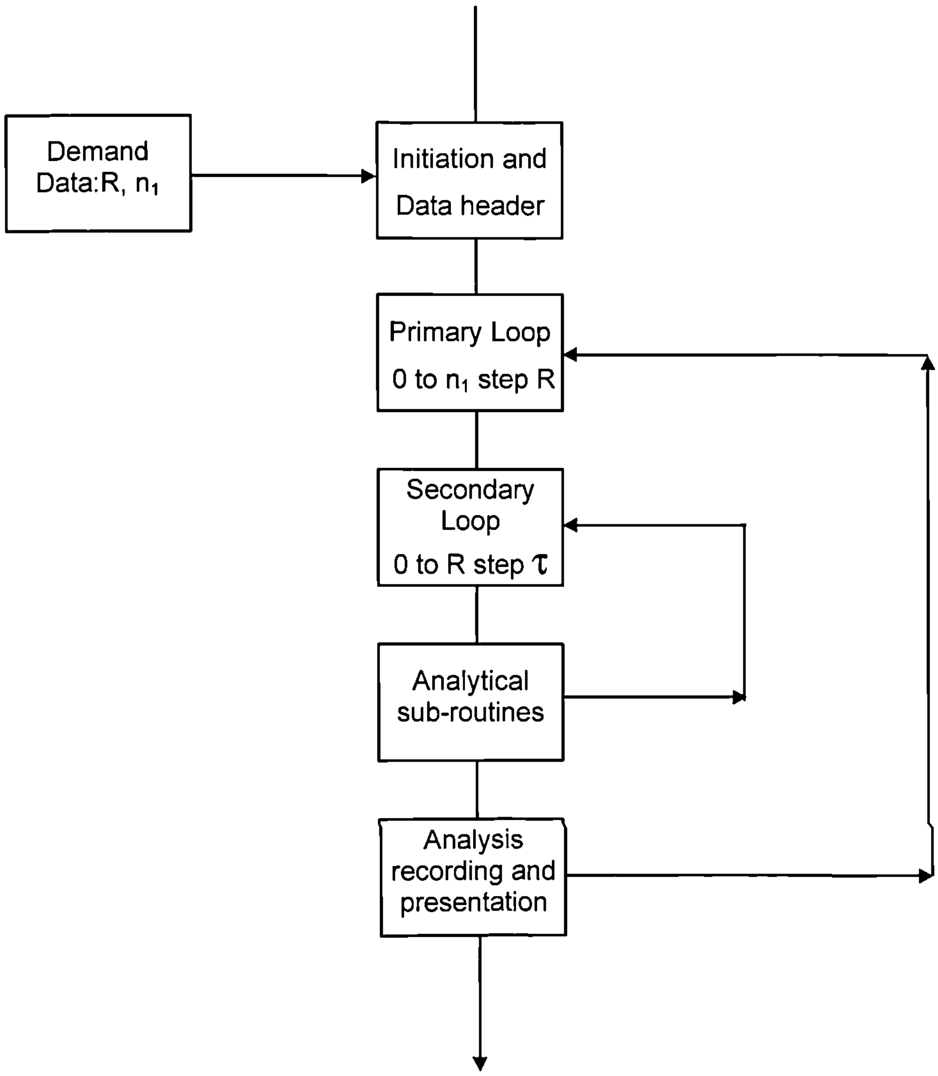


Figure 7.1 Model Overview

Figure 7.1 illustrates the relationship of analytical sub-routines to the data and simulation management sub-routine, while Figure 7.2 illustrates in broad terms the relationship between the main analytical sub-routines. Which sub-routines are called and in what sequence depends on the particular mode of operation being simulated (CHP/HP or CHP). This decision is taken by the user or by the model. Within the CHP/HP subroutine (designated *chphp_calc*) and the CHP subroutine (designated *chp_calc*), the operation of the relevant mode is simulated. Sub-routines pertaining to the characteristic of plant components (e.g. heat exchangers and electrical conversion efficiency) are also called, and the derivation of these common sub-routines will be discussed individually. After the plant operation has been simulated, relevant variables are passed to a sub-routine that assesses the economic and environmental performance of the simulated plant. Finally, a heat load sub-routine is called which calculates the return temperature of the LPW water system to the plant.

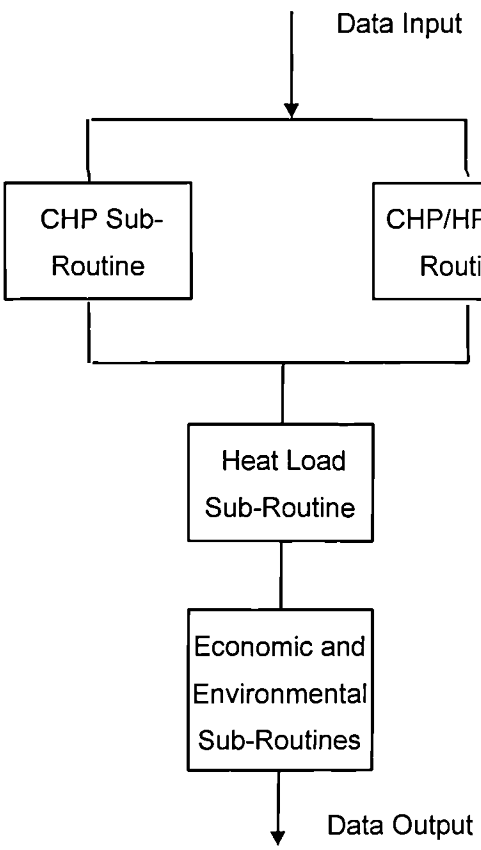


Figure 7.2 Analytical Sub-Routine Sequence.

7.4 Sub-Routine Structures

The following section will detail the derivation and structure of the individual sub-routines. As many sub-routines are called by others, the relationships between sub-routines will be justified.

7.4.1 CHP Simulation Sub-Routine (designated *chp_calc*)

The CHP sub-routine simulates the operation of the prototype plant operating in CHP mode. illustrates the structure of the sub-routine, which is listed in Appendix F.4. Function derivation and sub-routine structure will be described in the following section.

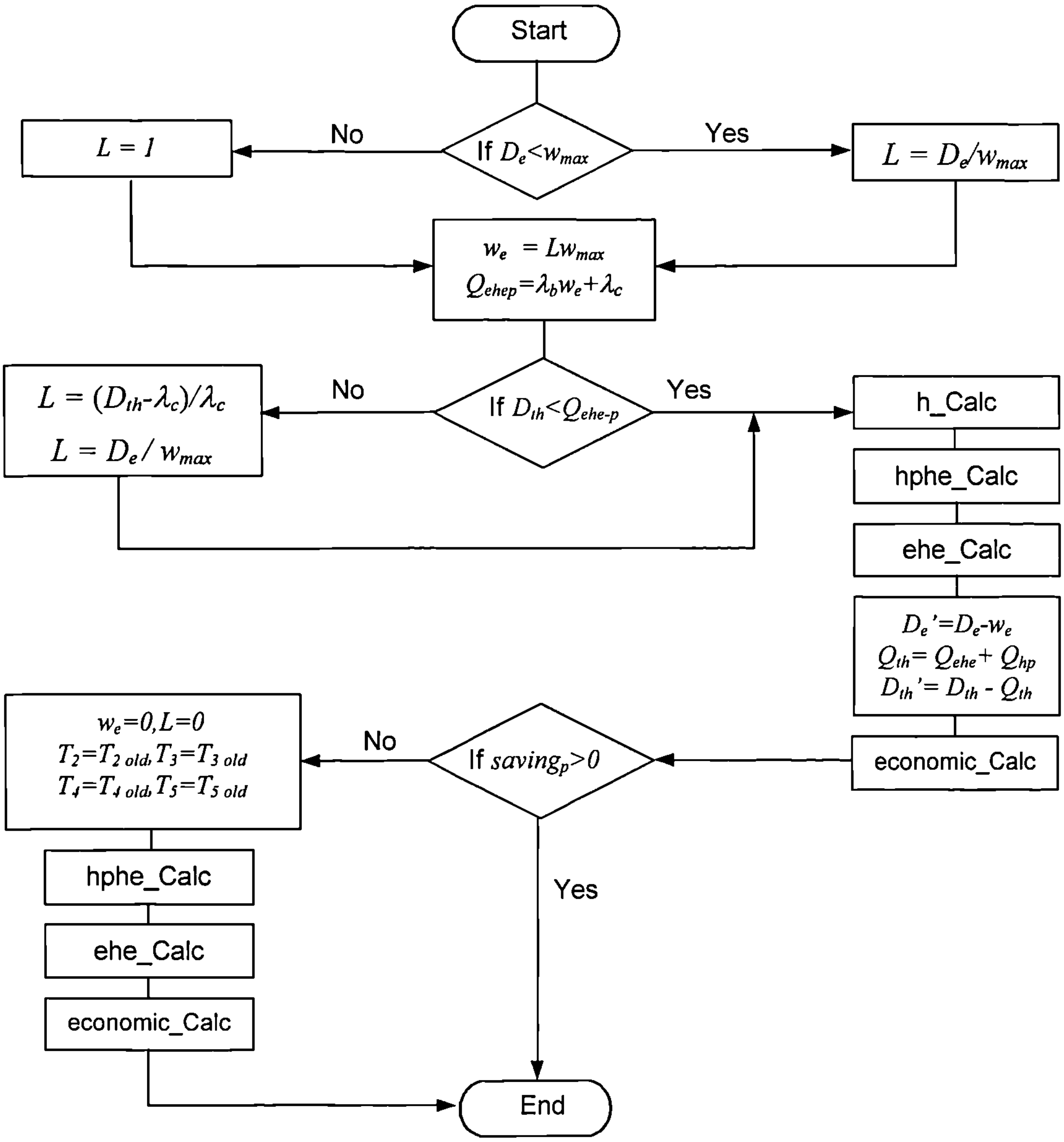


Figure 7.3 CHP Sub-Routine Structure

As the load factor (L) is calculated from:

$$L = \frac{D_e}{w_{\max}},$$

it is necessary to insert a condition to prevent the load factor being greater than 1. This would occur when the electrical demand is greater than the maximum electrical output of the generator (w_{max}). In this case, the value of the load factor (L) would be set to *one* by the conditional statement. Otherwise the above function is used to calculate the load factor (L).

After load factor has been established, the *potential* steady state thermal output of the EHE is calculated (Q_{ehe-p}). It is necessary to ignore the thermal capacitance effects of the EHE at this point in the analysis, as will be demonstrated later in this section. At this point, the thermal output of the EHE is calculated for steady state conditions from the linear relationship identified in Section 6.2.2. The values from the function gradient (λ_b) and the function y-intercept (λ_c) are set in the initialisation sub-routine of the model, to allow values to be altered for the evaluation of other designs of heat exchanger.

If the thermal output of the EHE exceeds the dwelling's thermal demand, then a conditional statement will recalculate load factor from:

$$L = \frac{w_e}{w_{\max}}$$

where w_e is calculated from:

$$w_e = \frac{D_{th} - \lambda_c}{\lambda_b}$$

Which is derived from rearranging:

$$D_{th} = \lambda_b w_e + \lambda_c$$

As $D_{th} = Q_{ehe}$, when the CHP mode satisfies thermal demand.

If the thermal output of the EHE is less than the thermal demand, the load factor (L) remains unchanged.

Subsequent to the calculation of load factor, the electrical conversion efficiency is calculated by calling the relevant sub-routine (*h_calc*, see Section 7.4.5), and hence the fuel input to the plant (Q_f) is calculated. Also the sub-routines that calculate the actual thermal outputs of both the heat pump heat exchanger (*Qhphe_calc*, see Section 7.4.4) and the EHE (*Qehe_calc*, see Section 7.4.3) are called, which take into account the thermal capacitance of the heat exchangers. Although the heat pump is not operating when the plant is running in CHP mode, the LPW system flows through the heat pump and hence its thermal response to changing system temperatures must be calculated. Once the thermal condition of the plant and electrical output have been calculated, the adjusted electrical and thermal demands are calculated. Then the sub-routine analysing the economic performance of the plant (*economic_calc* see Section 7.4.7) is called. If the potential steady state savings (*savings-p*) are less than zero, the CHP simulation sub-routine terminates and variables are passed on, until it is called by the main program loop during the next iteration.

If the potential steady state savings are less than zero, then the load factor (L), thermal outputs and electrical output (w_e) are set to zero, in order to prevent uneconomic operation. Within the EHE and heat pump heat exchanger sub-routines (*Qehe_calc* and *Qhphe_calc*), the water temperature values for the LPW system are recorded prior to calculation. These values are recalled and transient thermal calculations are repeated using the recalled values. This is to prevent model instability as thermal capacitance calculations are repeated twice in one iteration if uneconomic operation is invoked. This effectively halves the time step size of the thermal calculation relative to the rest of the model.

The decision to run the plant must be made using the potential steady state thermal output of the plant, as the actual thermal output of the plant will be initially zero, due to the thermal capacitance of the heat exchanger. Hence if the decision to operate the plant was made using thermal output, taking into account thermal transients, the economic analysis would always show a negative saving and the plant would be prevented from running (this has implications for actual plant control, see Section 9.4.1).

7.4.2 CHP/HP Sub-Routine (designated *chphp_calc*)

The CHP/HP sub-routine simulates the prototype plant while operating in hybrid CHP/HP mode. The CHP/HP subroutine additionally simulates the prototype plant operating in heating only mode. See Appendix F.5 for code listings.

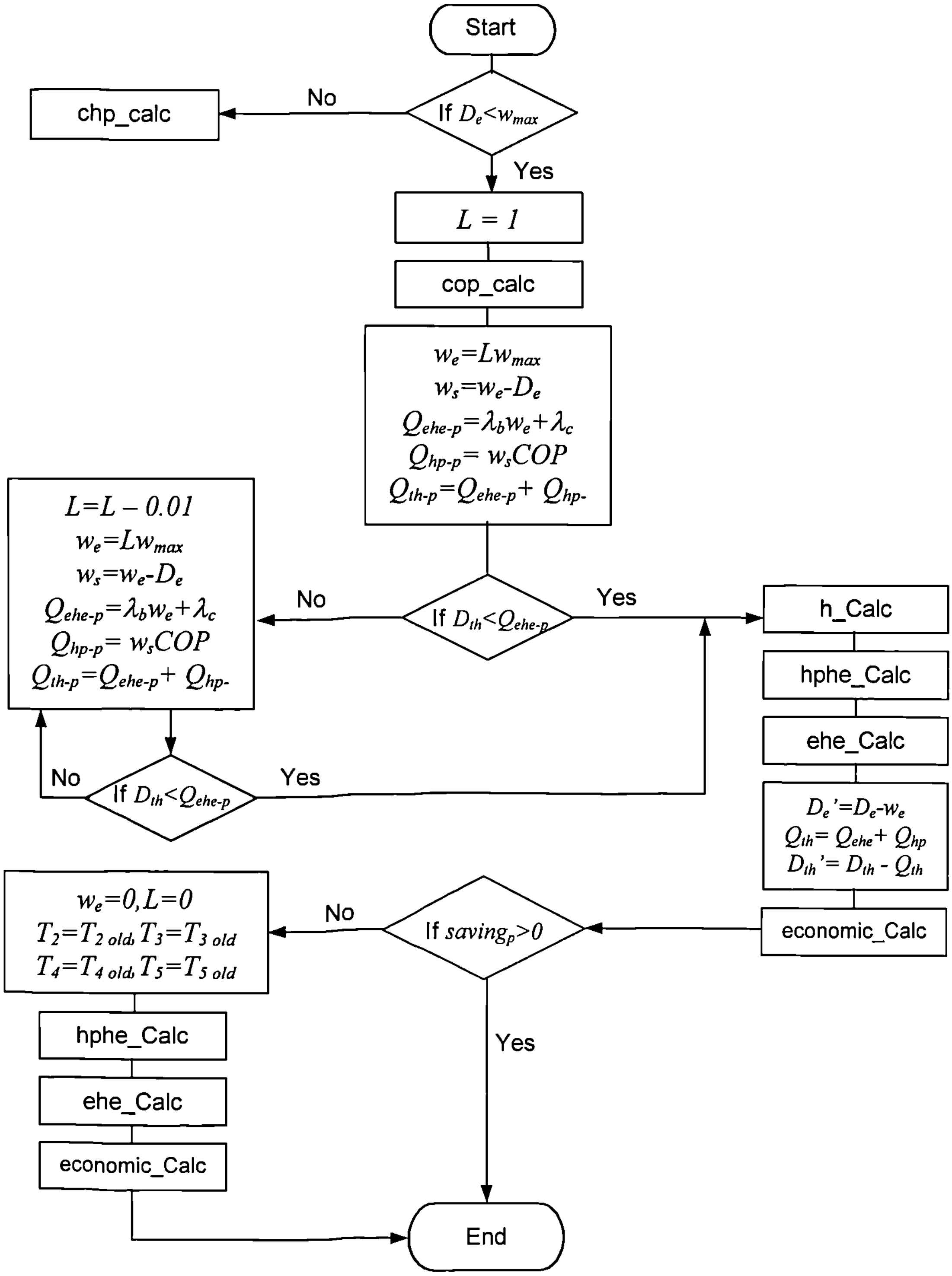


Figure 7.4. CHP/HP Sub-Routine

Figure 7.4 details the structure of this sub-routine, with the description following the same format as Section 7.4.1. On initiating the CHP/HP sub-routine, it ascertains if there is any spare generating capacity, i.e. if:

$D_e > w_{max}$

If no surplus capacity is available, the plant will run in CHP mode. To simulate this, a conditional statement terminates the CHP/HP sub-routine and calls the CHP sub-routine. If surplus generating capacity is available, then the sub-routine will set the load factor (L) to 100% i.e.: $L = 1$.

It is assumed initially that the engine/generator will primarily run at its full rated value in CHP/HP mode. The surplus electrical generation is calculated and the COP calculation sub-routine (*cop_calc,* see Section 7.4.6) is called to ascertain the COP of the heat pump for that surplus generation availability (w_s).

Potential steady state thermal outputs are calculated for both the EHE and heat pump. It is necessary to calculate potential steady state outputs, for the reasons discussed in Section 7.4.1. Total plant potential thermal output is then calculated.

If the total potential thermal output from the plant is greater than the requirement, the engine/generator output must be reduced to match both thermal and electrical demands. To ascertain the appropriate load factor, an iteration is carried out in which the load factor is reduced incrementally, i.e.: $L = L - 0.01$

The potential thermal output from the EHE and heat pump is recalculated for each load factor increment until the thermal and electrical demands are matched. After the required load factor has been ascertained, the structure of the sub-routine is identical to that for the CHP simulation sub-routine (see Section 7.4.1).

7.4.3 Exhaust Heat Exchanger Sub-Routine (designated *Qthehe_calc*)

As discussed earlier in this chapter, the evaluation model takes into account the thermal capacitance of both the EHE and heat pump heat exchanger. The following section will derive the mechanism for calculating the EHE thermal output, when thermal capacitance is taken into account, and describes the resulting structure of the EHE sub-routine (as listed in Appendix F.6).

The calculation of the EHE thermal output assumes:

- The water contained within the heat exchanger is at a uniform temperature.
- The heating effect of the exhaust gas is uniform.
- The density of water remains at 1kg/litre for the operating temperature range of the EHE (15°C to 100°C).

A simple time based finite difference method was developed.

Consider the EHE to have a fixed volume (V_{ehe}) and bulk average water temperature (T_{bulk}). Water enters and exits the EHE at the flow rate of the LPW system (m_{wt}) at temperature T_4 and T_5 respectively. With no heat input, see Figure 7.5 ($Q_{ex} = 0$), then:

$$T_4 = T_5 = T_{bulk}$$

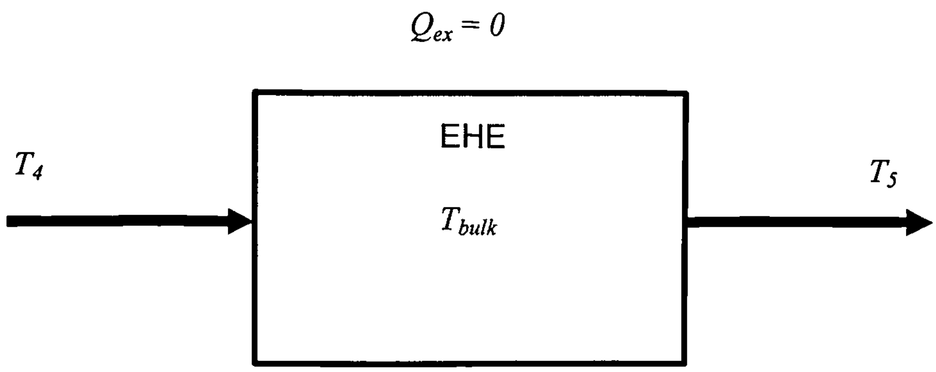


Figure 7.5. EHE at Initial State

Now consider $T_4 \neq T_{bulk} \quad T_5$

At t = 0, let the bulk temperature be: $T_{bulk}^{t=0}$.

Water at temperature T_4 will replace water leaving the heat exchanger at temperature T_{bulk}, at time interval 1: $T_5 = T_{bulk}$. The new bulk water temperature within the heat exchanger will be the average of T_4 and T_{bulk} with respect to their relative volumes (see Figure 7.6). The volume of water entering the EHE is m_w per unit time (τ). The volume of water, at T_{bulk}, remaining within the heat exchanger (assuming the density of water is one kg/Litre) is:

$$V_{ehe} - m_{wt}\tau$$

The total volume within the EHE remains constant at V_{ehe} as:

$$V_{ehe} = V_{ehe} - m_{wt}\tau + m_{wt}\tau$$

Hence the new average bulk temperature, at interval time t = 1 is:

$$T_{bulk}^{t=1} = \frac{T_4 m_{wt}\tau + T_{bulk}^{t=0}(V_{ehe} - m_{wt}\tau)}{V_{ehe}}$$

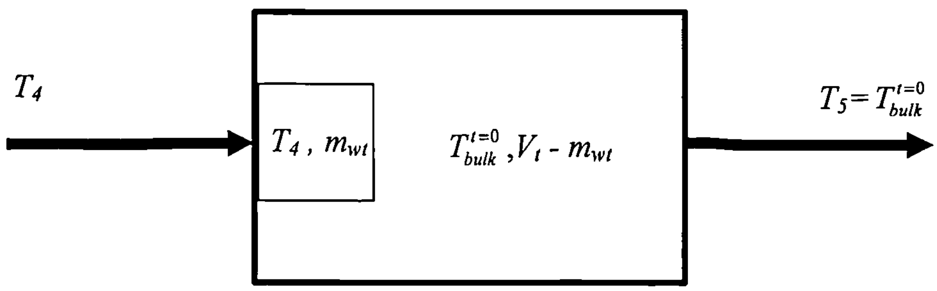


Figure 7.6. EHE at t = 1

Now consider a heat input Q_{ex} per unit time. An increase in bulk water temperature, due to the thermal input (Q_{ex}) per unit time, (τ) is calculated by rearranging:

$$Q_{ex} = \frac{V_{ehe}\rho_{tw} cp_{wt} (T_{bulk}^{t=1} - T_{bulk}^{t=0})}{\tau}$$

To give:

$$T_{bulk}^{t=1} = T_{bulk}^{t=0} + \frac{Q_{ex}\tau}{V_{ehe}\rho_{wt} cp_{wt}}$$

For the next time interval (t = 2), the temperature of the water leaving the EHE (T_5) will be that of the bulk temperature for the previous time interval, $T_{bulk}^{t=1}$. This procedure is summarised in Figure 7.7.

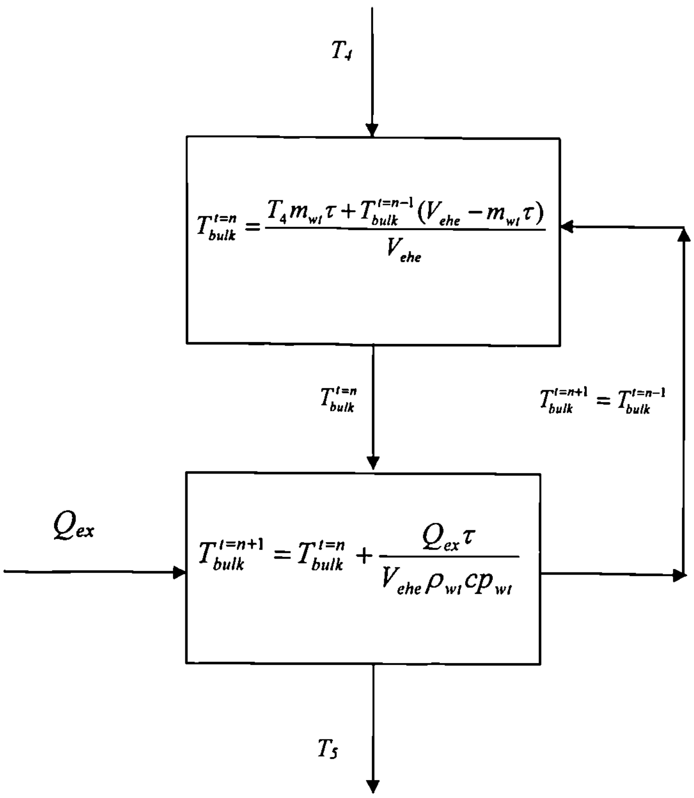


Figure 7.7 Heat Exchanger Thermal Capacitance Calculation Procedure.

The previously described procedure was used in the EHE simulation sub-routine, as illustrated in Figure 7.8.

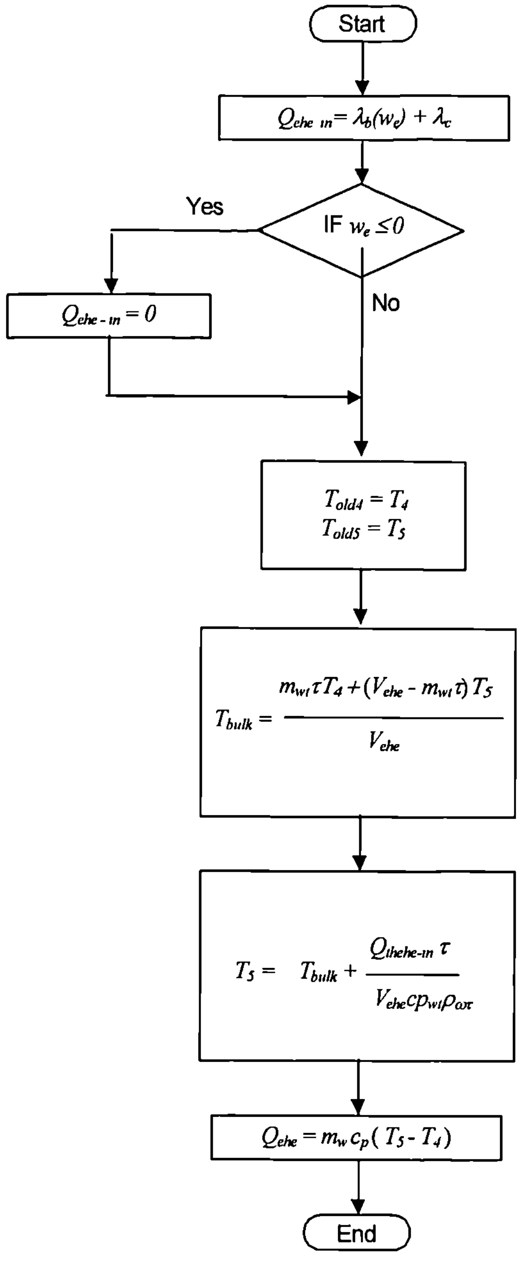


Figure 7.8. Structure of EHE Simulation Sub-routine.

On initiating, the sub-routine calculated the steady state heat delivery of the EHE from:

$$Q_{thehe_{in}} = \lambda_b w_e + \lambda_C$$

It is assumed that the steady state output of the EHE is equivalent to the heat delivery from the exhaust gas (to the EHE). In making this assumption, losses due to thermal bridging and convective losses from the EHE body can be accounted for. Although the actual heat input will be higher, some fraction will be lost through convection and conduction via the EHE body. The thermal delivery to the water represents the exhaust gas thermal delivery to the EHE, once these losses have been experienced. The constants are derived from experimental results (see Section 6.2.2).

As the thermal delivery to the EHE is assumed to have a linear relationship to engine load, at zero load the thermal delivery would be:

$$Q_{thehe_{in}} = \lambda_b(0) + \lambda_C = \lambda_C$$

Hence a conditional statement must be added to the sub-routine, to set the thermal delivery to zero when the engine is not running.

Prior to water temperature calculation, the water temperatures are recorded and stored as 'old' values. This is to avoid model instability problems discussed in Section 7.4.1.

The bulk water temperature and output water temperature are calculated using the previously described procedures. Finally, the heat delivery to the LPW system is calculated by considering the input and output LPW system temperature and flow rate.

The previously used notation for bulk water temperature, denoting time interval, is not used in Figure 7.8, as this notation was not utilised in the model and is only used in the derivation of thermal capacitance calculations.

7.4.4 Heat Pump Heat Exchanger Sub-Routine (designated *Qthhp_calc*)

The derivation of the heat pump LPW system heat exchanger sub-routine is identical to that for the EHE (see Section 7.4.3). **Figure 7.9** describes the structure of this sub-routine and code listings are contained in Appendix F.7. On initiating, a conditional statement prevents a negative value for heat pump thermal delivery from being calculated. If surplus generating requirement exists, the thermal delivery from the heat pump is calculated from:

$$Q_{hp_{in}} = w_s COP$$

The amount of surplus generating capacity is calculated in the CHP/HP simulation sub-routine (see Section 7.4.2). The COP of the heat pump is calculated within the *heat pump performance* sub-routine (see Section 7.4.6), which is called by the CHP/HP sub-routine.

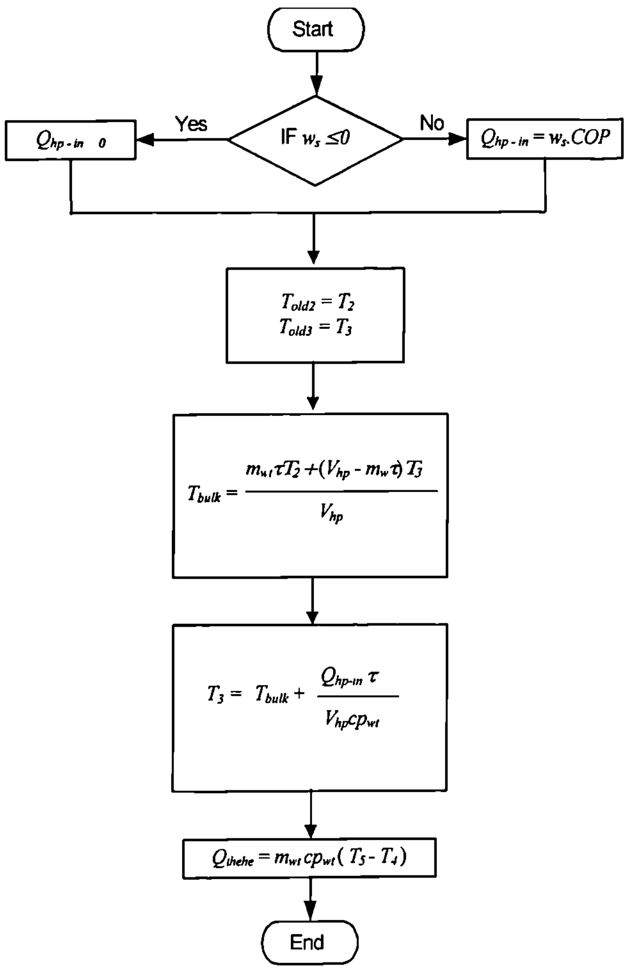


Figure 7.9 Heat Pump Simulation Sub-Routine Structure.

7.4.5 **Electrical Conversion Efficiency Sub-Routine** (designated *h_calc*)

In order to find the fuel consumption of the engine and hence of the plant, the *electrical conversion efficiency* sub-routine uses experimental data to calculate electrical conversion efficiency. Experimental results were used to derive a polynumeric expression (see Section 6.2.1), which represents the relationship between engine/generator load and electrical conversion efficiency. The resulting function is then used to find the electrical conversion efficiency for any given engine load and hence fuel consumption. Within the *h_calc* sub-routine (see Appendix F.8 for code listings), the polynumeric function is of a general form:

$$\eta_e = A_{\eta_e} w_e^{\,3} + B_{\eta_e} w_e^{\,2} + C_{\eta_e} w_e + D_{\eta_e}$$

The values for constants are held in the initiating sub-routine and are loaded when the model is started. The fuel consumption is subsequently found from:

$$Q_f = \frac{w_e}{\eta_e}$$

7.4.6 **Heat Pump Performance Sub-Routine** (designated *cop_calc*)

The *heat pump performance* sub-routine calculates the COP of the heat pump. As discussed in previous sections, the plant performance is very sensitive to heat pump COP. In order to assess different designs and configurations of heat pumps, the *cop_calc* sub-routine was constructed to model a number of different potential heat pump performance characteristics (see Appendix F.9 for code listings):

- An experimentally based relationship, with an approximately linear COP, above 0.87kW of electrical input.
- A parabolic relationship (between electrical input and COP) based on a *by-pass* modification of the *Versatemp* unit.
- A generalised exponential relationship that can be varied to give an infinite number of relationships, to allow for the modelling and optimisation of different refrigerant systems.

The performance characteristic to be used is defined by the user. Each of the options will be individually discussed. Figure 7.10 illustrates the overall structure of the sub-routine.

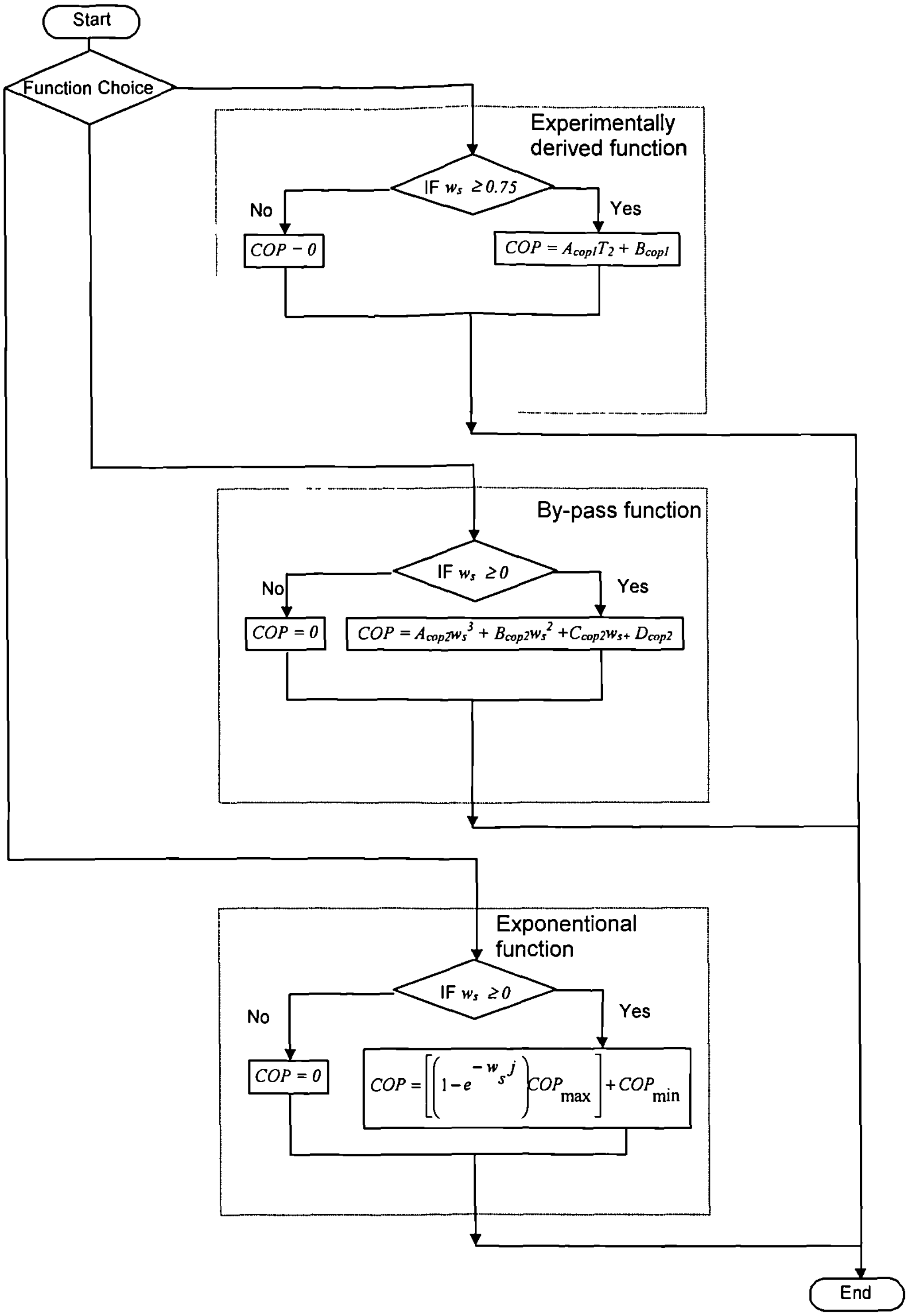


Figure 7.10 COP Calculation Procedure

7.4.6.1 Experimentally Based Heat Pump Simulation

With respect to Figure 7.10, the unmodified *Versatemp* system would only operate when the surplus power availability was above 0.87kW (see Section 6.2.3) and hence a conditional statement was required.

7.4.6.2 By-Pass Heat Pump Performance

A separate model was written to simulate the *Versatemp* heat pump, with an evaporator/condenser by-pass modification. As the *Versatemp by-pass* model is itself complex, it was inappropriate to incorporate this in the *plant evaluation model,* as this would considerably retard simulation speed and add complexity. The results of the model were used to derive a polynumeric function, which is used within the *cop_calc* sub-routine, of the generalised form:

$$COP = A_{cop_2} w_s^{\,3} + B_{cop_2} w_s^{\,2} + C_{cop_2} w_s + D_{cop_2}$$

The values for the constants are held in the initialisation sub-routine. A conditional statement was also required to force the COP to zero when the work input to the heat pump is zero.

7.4.6.3 Generalised Exponential COP Characteristic Function

To generate a COP characteristic to represent varying COP/work input relationships, a generalised exponential function was used:

$$COP = \left[\left(1 - e^{-w_s j}\right)COP_{\max}\right] + COP_{\min}$$

Where (with reference to Figure 7.11):

$COP_{\max}$ is the maximum obtainable COP that the heat pump is allowed to reach.

$COP_{\min}$ is the minimum COP of the heat pump when $w_s \rightarrow 0$, or the y-intercept of the function.

j is the exponential constant of the function. Altering the value of j will modify the value of w_s at which COP reaches the maximum obtainable value, as illustrated in Figure 7.11.

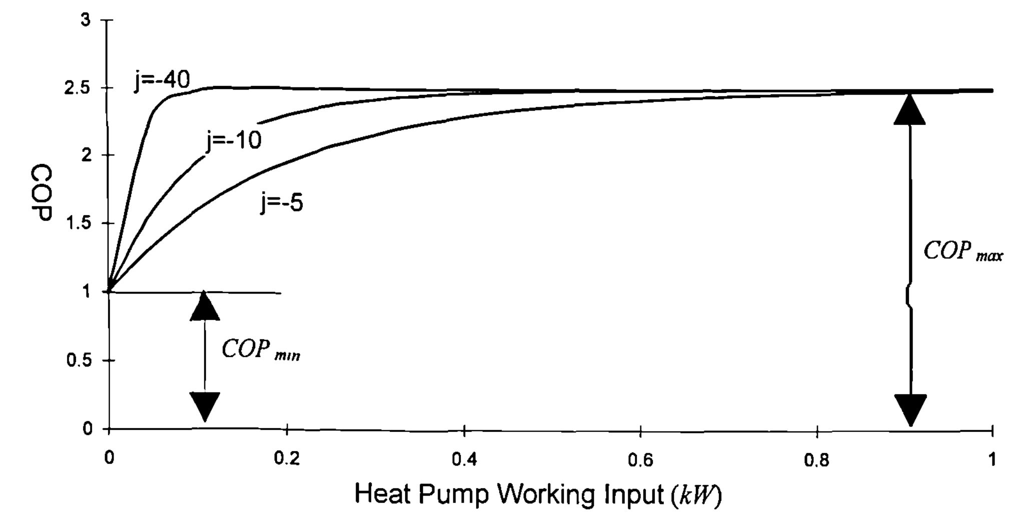


Figure 7.11 Generalised Exponential COP/w_s Relationship

7.4.7 Economic Evaluation Sub-Routine (designated *economic_calc*)

The *economic evaluation* sub-routine, when called by either *chp_calc* or *chphp-_calc,* calculates the running costs for the plant and the equivalent running costs for conventional domestic energy costs in meeting the same energy output. The two values are used to calculate the financial savings.

The running costs of the plant (c_r) are found by taking into consideration fuel unit cost (c_f) and plant maintenance unit cost (c_m) which are expressed in costs per unit generated electricity (at the generator and not delivered to the dwelling):

$$c_r = Q_f c_f + w_e c_m$$

The conventional costs for meeting the energy output consider energy unit costs and conventional boiler efficiencies. When calculating savings, both steady state and transient thermal deliveries are used, to give 'actual' savings and potential steady state savings. The use of potential steady state savings is necessary within the model for the reasons given in Section 7.4.1.

Potential steady state savings are calculated from:

$$savings_p = \left(w_d c_e + \frac{Q_{th_p} c_f}{\eta_{boiler}} \right) - c_r$$

and for 'actual' savings:

$$savings = \left(w_d c_e + \frac{Q_{th} c_f}{\eta_{boilerr}} \right) - c_r$$

The cost of meeting additional energy requirements, when demands are greater than plant capacities, from conventional sources is not considered. As a conventionally supplied dwelling would also have to incur this additional cost, it would not affect the relative cost effectiveness of the plant. Code listings for the *economic_calc* sub-routine are contained in Appendix F.10.

7.4.8 Environmental Evaluation Sub-Routine (designated *enviro_calc*)

The *environmental evaluation sub-routine* is used to calculate the carbon dioxide emissions evolved as a consequence of the energy demand of the subject dwelling. Carbon dioxide emissions are calculated for both conventional energy supply and the domestic co-generation mode being modelled - allowing for comparative environmental analysis. The method of calculating carbon dioxide emissions is similar to that employed in Section 3.6.1 and that used in previous domestic scale CHP studies [24]. Appendix C contains detailed derivations of the environmental analysis and associated constants and code listings for *enviro-_calc* are held in Appendix F.11.

7.5 Model Validation

The following sections will describe the validation exercise carried out on the *concept evaluation model*, as developed in Sections 7.1 to 7.4. The aim and necessity of the validation process will be set out. Comparative techniques and analytical methods employed are described prior to the presentation and analysis of the actual validation exercise, which is given with a tabulated summary and statement of model validity.

7.5.1 Validation Requirements and Aims

It is necessary to rigorously test the model results against those obtained experimentally, to ensure that any analysis resulting from subsequent simulations is reliable. By carrying out a thorough validation process, a statistical confidence can be assigned to the model results. Additionally, the validation process will:

- Serve as an additional test of the structure of the model.
- Assess the experimental methodology and data acquisition system.
- Complement the analysis of experimental results (in Chapter 6). Test the assumptions made in Section 7.2 and identify any additional processes not considered by the model.

7.5.2 Validation Methods

As the concept evaluation model is intended to examine the economic and environmental performance of the CHP/HP concept, it is necessary to validate the model results which have direct influence on plant performance (i.e. critical parameters). The critical parameters are:

- Fuel consumption.
- Thermal outputs (both from individual heat exchangers and the total output).

Many other parameters could be validated. However, these are associated with the calculation of the critical parameters and hence are implicitly considered.

The general principle of model validation was to configure the model to simulate a test of the experimental plant and then to compare the critical parameters at each time interval. By this method, both transient and steady state effects could be observed.

7.5.2.1 Validation Test Data and Model Configuration

The concept evaluation model is 'driven' by electrical demand, as the engine load is used to calculate the critical parameters (see Section 7.3). The recorded electrical output of the actual prototype plant can therefore be used as the electrical demand required for the model. By using this actual plant electrical output, the model should calculate other plant conditions independently and hence a complete test run can be simulated, by changing the electrical demand at the appropriate point. Experimental tests were selected for validation use on the grounds of transducer reliability and range of operating conditions.

Other experimental results required were the return temperature and flow rate of the LPW system. The flow temperature from the heat exchangers was calculated by the model using the experimental data. The direct use of experimentally obtained heat exchanger flow temperatures would negate many model functions and hence render the validation process meaningless.

Validation data, used to 'drive' the model, were loaded into the model using the external database, which contains the demand profiles (see Section 7.3). Filtered experimental data were used (see Section 5.11). As the time interval of the experimental data was variable, it was also necessary to alter the time interval on the *run_loop* for each main loop interval. The time base pertaining to a particular data point was loaded into the model with the driver data.

The thermal demand on the plant was set to a constant high value, to prevent the model from reducing engine load due to low thermal demand.

7.5.2.2 Analytical Methodology

Comparison of simulated and actual results on a time basis will allow for the identification and subsequent analysis of any disagreement between the sets of results. The analysis of disagreements will be used to develop correction factors, to be used within the model. After any alteration to the model, the relevant validation simulation will be re-run.

Simulated and actual data sets for each critical parameter were correlated and the following statistical indicators were calculated to assess the accuracy of the model with respect to the relevant parameter.

Regression Constants and Limits:

Carrying out a regression on the correlated data will yield the constants of a straight line function, i.e. $y = \beta_0 + \beta_1 x$

For a perfect model (where the simulated results match the actual results):

$\beta_0 = 0$ and

$\beta_1 = 1$.

Upper and lower limits will also be prescribed to the regression constant to a 95% confidence interval, i.e. there will be 95% probability that the reported constants are within the respective upper and lower limits.

Coefficient of Determination (r^2) :

Coefficient of Determination (r^2) is an indicator of the strength of a linear relationship. A result of over 0.6 is generally considered to indicate a strong relationship between the correlated values and hence a good model. A value of 1 would indicate a perfect correlation and hence a perfect model.

t-Statistic Test:

By comparing the t-statistic for the regression to the relevant *student t test* for the result's population size, the unity of the model can be assessed. If the t-statistic is greater than the student t test value for a given confidence interval (in this case 95%), then it indicates that the relationship between data sets is statistically valid and hence the model is acceptable.

The t-statistic will only be carried out for the gradient component (β_1) of the regression function, as the intercept component is negligible and can be generally ignored.

7.5.3 Validation of Critical Parameters

In the following sections, the validation analysis will be carried out for the fuel consumption and thermal output parameters listed in Section 7.5.2. Full reference will made to the relevant test data where applicable. Figures relating to the following sections can be found at the end of this chapter.

7.5.3.1 Validation of Fuel Consumption (Q_f)

The simulated and actual fuel consumption data for the duration of *CHP Test 1* (see Appendix E.1.1) are shown in Figure 7.15. In this test, the engine was initially run at full load and then for decreasing loads. By inspection, the simulated and actual results give good agreement except for the initial start up period and at the final stages of the test. The latter anomaly can be attributed to power sensor tolerance. The initial anomaly cannot be purely explained by transducer uncertainty: it is likely that the data recording procedure was interrupted during this period, giving a false reading.

Not withstanding the two anomalies identified above, the correlation of the two sets of results indicates a reliable simulation (see Figure 7.16). With reference to Table 7.1, the r^2 value indicates a strong correlation and the t-statistic is considerably higher than its critical value. Hence the simulation of fuel consumption can be considered valid. The value of the correlation coefficient also adds validity to the simulation with a negligible intercept (β_o) and a gradient (β_1) approaching unity.

As the calculation of fuel consumption in the concept evaluation model requires only simple functions, the validation of this parameter tests the model code structure and the empirical data in the *h_calc* sub-routine.

Table 7.1. Fuel Consumption Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.89	0.6
t-Static	14.46	2.08
β_0	0.32 (±0.705)	
β_1	0.975 (±0.14)	

7.5.3.2 EHE Thermal Delivery (Q_{ehe})

The simulation of the EHE was considerably more complex than that of fuel consumption, hence the validation of the Q_{ehe} parameter will not only test model syntax and empirical reference data but also the assumptions and analytical approach adopted in the development of the *Qehe_calc* sub-routine (as described in Section 7.4.3). In order to carry out a rigorous test, two sets of experimental tests were simulated - the *CHP Test 1* (see Appendix E.1.1) and *CHP Test 2* (see Appendix E.1.5). The initial test (using *CHP Test 1*) will test the model's reaction to a rapidly changing engine load, while the later test (using *CHP Test 2*) will examine the simulation of a steady engine load.

7.5.3.2.1 Rapid Transient Response

Inspection of Figure 7.17 indicates good general agreement between simulated and actual EHE thermal delivery. Although the simulated results follow the general trend of the actual results, a number of anomalies exist, in particular at time 28 minutes and 1h 15m.

These anomalies could be attributed to transducer uncertainty, however their nature is consistent with model resolution effects. As the driver data is loaded into the model at a specific (and often varying) time base, the *Qehe_calc* sub-routine will calculate the flow temperature from the EHE with respect to the last value EHE return temperature (T_4) imported. The model will use this data until the next data set is loaded. If the actual value of T_4 should rapidly change between time intervals, the effect will be to cause a rapid change in Q_{ehe}, as the flow temperature (T_5) will still largely reflect the previous value of T_4 (as a consequence of the EHE thermal capacitance calculation). In reality, the return temperature T_4 will vary gradually. However, owing to the resolution of data in the model, it appears to change instantaneously and hence the observed anomalies are the model's response to the apparent rapid temperature variation. This problem could be overcome by using data of a higher resolution. In this case, second by second experimental data could be used, although the use of un-filtered data would cause additional problems. No action was taken to eliminate these anomalies, as the return temperature will be kept constant during the concept evaluation phase (see Chapter 8). Also, despite these anomalies, the simulation of this parameter is statistically satisfactory, as is demonstrated below.

With reference to Figure 7.18, the correlation between the simulated and actual data indicates a strong agreement - the anomalous results discussed previously are visible as rogue points. All the statistical indicators report satisfactory results, presented in Table 7.2. Therefore the simulation of rapid thermal transience is valid and hence the assumption and modelling techniques employed in Section 7.4.3 are correct.

Table 7.2. Rapid Thermal Transient Response Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.82	0.6
t-Static	10.63	2.08
β_0	0.07 (±0.45)	
β_1	0.97(±0.19)	

7.5.3.2.2 Transient Response to Steady Engine Load and to Shut Down

CHP Test 2 (see Appendix E.1.5) was carried out at full load (from cold) until the plant was shut down. In this test, the data will be used to examine the simulation of long term thermal transience (compared to rapid transient response).

Figure 7.19 compares the actual and simulated EHE thermal delivery (Q_{ehe}). Reasonable agreement can be observed between actual and simulated results. The cooling period (from time 22 minutes) shows extremely good agreement. It can therefore be inferred that the thermal capacitance calculation procedure (see Section 7.4.3) is valid. Hence the assumptions, that only thermal capacitance of the water within the EHE need be considered and that the conductive losses from the EHE to the rest of the plant were negligible, are correct for steady state and cooling periods. In contrast, the initial simulated thermal response does not agree with the actual response, to the same extent as for the cooling period. The simulation consistently over-estimates the thermal delivery. This over-estimation can be attributed to:

- Error in empirical model data - this is unlikely given the results of the previous comparison (Section 7.5.3.1).
- Conductive losses from the EHE.

During the initial period of the test (with a 'cold' plant) the EHE would experience conductive losses, as the EHE would heat up faster than the rest of the plant. Additionally, during the initial start up period, the 'cold' structure of the EHE effectively increases the EHE's relative thermal capacitance. To correct this error, a correction factor could be placed in the model to either adjust the plant's thermal mass or the exhaust heat delivery during the initial stages of the run. However, any such modification would introduce further uncertainty into the model, which despite this anomaly still performs well - as the following analysis will show.

Correlation analysis (see Figure 7.20) confirms the model validity despite the previously described anomaly. See Table 7.3 for tabulated results. Very high values for r^2 and the t-statistic are returned, hence the simulation of Q_{ehe} (for CHP operation) can be assumed to be valid. The regression coefficients reinforce the validity of the model.

Table 7.3. Steady Engine Load and Shut Down Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.97	0.6
t-Static	30.08	2.08
β_0	-0.19 (±0.11)	
β_1	0.95(±0.058)	

7.5.3.3 Validation of Heat Pump Thermal Delivery (Q_{hp})

In order to assess the model's performance with respect to the simulation of the heat pump thermal delivery (Q_{hp}), the experimental test *CHPHP Test 2* (see Appendix E.1.6) was used. In this test, both CHP and heat pump components were in operation. The plant was started up from cold and ran at full load. At a given point, the heat pump was switched off and allowed to cool prior to restarting (see Figure 7.12). Using this particular test gives a good range of heat pump running conditions, cold start up, hot start up and shut down.

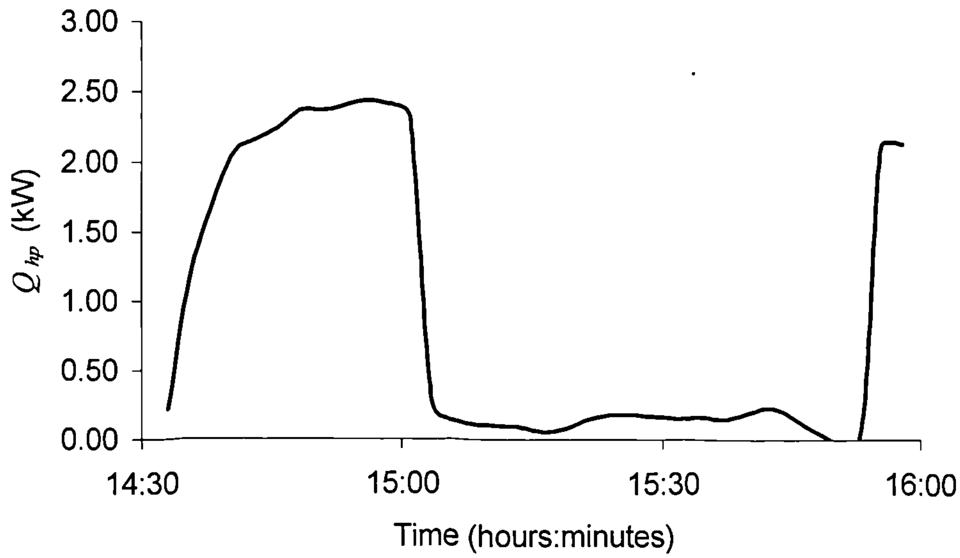


Figure 7.12. Heat Pump Duty during *CHPHP Test 2*

Figure 7.21 compares the actual and simulated thermal output of the heat pump LPW system heat exchanger (Q_{hp}). This was calculated by the *Qhp_calc* sub-routine (see Section 7.4.4), employing the same procedure used in the calculation of EHE thermal output and utilising similar modelling assumptions.

Inspection of Figure 7.21 shows very high agreement for the shut down thermal transient stage and for the 'hot' start up period. This close agreement infers that the thermal transient calculation technique and empirical model data are correct for a 'hot' heat pump. As with the simulation of the EHE thermal delivery, the model over-estimates the thermal delivery of the heat pump heat exchanger during the initial 'cold' start up period. This mismatch can again be attributed to modelling assumptions: however, in this case the anomaly is extremely pronounced. It must be concluded that the heat pump heat exchanger experiences relatively large conductive losses. Unlike the EHE, which is a discrete well-insulated component (with minimal surface contact with the rest of the plant), the heat pump LPW system heat exchanger is in contact with many other heat pump components. Additionally, the heat pump compressor will increase the thermal mass of the heat pump unit until it obtains a steady state running temperature (see Section 6.2.3). These effects create high heat pump heat exchanger losses during a 'cold' start up. This mismatch must be addressed, as unlike the similar anomaly observed in the EHE simulation, the model validity is compromised by the mismatch.

Table 7.4 presents the statistical indicators for the *CHPHP Test 2* actual/ simulated results correlation, as shown in Graph 7.22. All the indicators, for heat pump thermal delivery (Q_{hp}), show a satisfactory result. However, as the model is simulating CHP/HP mode, the total thermal delivery from the plant must be examined. Figure 7.23 presents a comparison between the simulated and actual results for the combined CHP and heat pump thermal delivery (Q_{th}). Correlation (see Figure 7.24) and subsequent analysis (see Table 7.5) show that the model is unreliable, with a very low r^2 value. As the total thermal output is a sum of the EHE and heat pump thermal deliveries, their individual errors, which are individually acceptable, are combined. The combination of errors renders the simulation of the total thermal output unacceptable. Hence the 'cold' start up mismatch must be addressed.

Table 7.4. Heat Pump Delivery Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.91	0.6
t-Static	11.47	2.08
β_0	0.12 (±0.22)	
β_1	0.81(±0.14)	

Table 7.5. Total CHP/HP Thermal Delivery Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.55	0.6
t-Static	5.08	2.08
β_0	0.92 (±0.86)	
β_1	0.70(±0.7)	

Inspection of the 'cold' start up anomaly (see Graph 7.23) shows an initial 2.3kW mismatch (for Q_{th}): this will be considered to be the effective loss from the heat pump. This loss attenuates as the heat pump compressor nears thermal equilibrium, at which point the actual thermal delivery matches the simulated value (see Figure 7.13). Figure 6.5 shows the thermal transience of the heat pump compressor (for the same experimental test). It can be concluded that the effective heat loss has an exponential characteristic, which will tend to zero when compressor thermal equilibrium is obtained. This assumption was used to develop a correction function for the model, which is of an exponential form (see Figure 7.14) and is called only for a 'cold' start up. The correction factor calculates the effective heat loss and reduces the heat pump thermal delivery accordingly.

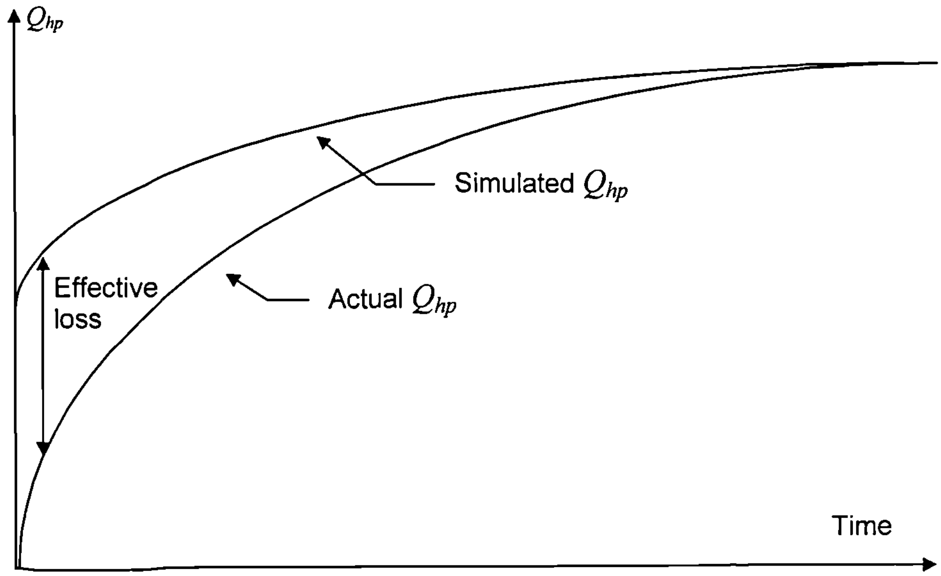


Figure 7.13 Effective Heat Pump Thermal Loss

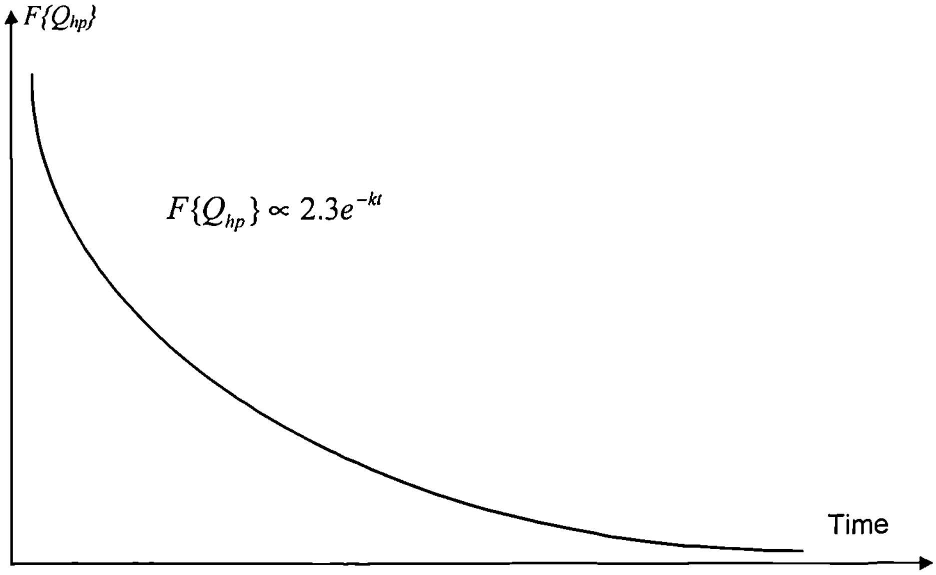


Figure 7.14 Heat Pump Thermal Correction Factor

The heat pump thermal correction factor was inserted into the concept evaluation model (see Appendix F.2). A conditional statement only allows the heat pump thermal correction to be used during the start up periods, when the heat pump has not been used for a number of hours.

Graph 7.25 shows the modelled result for heat pump thermal delivery (Q_{hp}), when the heat pump thermal correction factor is utilised for an initial start up period. It can concluded that the correction factor provides a satisfactory simulation of heat pump thermal delivery, given the close agreement evident from the associated correlation (see Figure 7.26) and statistical analysis (see Table 7.6). It must be noted that the particularly high value of r^2 for the heat pump thermal delivery is to be expected, given the empirical nature of the correction factor.

The combined CHP/HP thermal output (Q_{th}) performs satisfactorily when the heat pump correction factor is employed. With reference to Figure 7.27, the simulated and actual combined thermal outputs now exhibit good agreement. The resulting regression analysis of Figure 7.28 (see

Table 7.7) shows that the simulation of the combined plant thermal output is valid. Combined thermal output returns a reliable value for r^2.

Table 7.6. Adjusted Heat Pump Delivery Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.99	0.6
t-Static	57.00	2.08
β_0	0.11 (±0.05)	
β_1	0.96(±0.03)	

Table 7.7. Adjusted Total CHP/HP Thermal Delivery Validation Indicators

Indicator	Value	Satisfactory Value
r^2	0.85	0.6
t-Static	12.37	2.08
β_0	0.47 (±0.52)	
β_1	0.86(±0.14)	

7.5.4 Summary of Validation

It has been demonstrated that the simulation of critical parameters by the concept evaluation model is valid. All the simulated parameters performed satisfactorily under statistical analysis. Only when parameters were combined, giving rise to increased model uncertainty, did the model not perform reliably. The use of an empirically derived correction factor improved model reliability. Table 7.8 summarises the statistical analysis of all the validation tests.

Minor anomalies, which did not compromise model validity, were evident. These were due to experimental and data acquisition procedures. In order to filter transducer errors and 'noise' from the transducer signal, it was necessary to average data over a relatively long period, which gives rise to instability when data is subsequently used to drive the model. The structure of data acquisition software also contributed to extended data time periods.

The concept evaluation model obtained satisfactory performance according to statistical analysis and visibly gives good agreement with actual experimental results. Hence it will be assumed that the results of the following evaluation modelling of the CHP/HP concept (in Chapter 8) are valid within the statistical limits.

Table 7.8. Validation Statistical Analysis Summary

Parameter	Description	Test	r^2	T-Stat	Critical t
Q_f	Fuel Consumption	CHP Test 1	0.89	14.46	2.08
Q_{ehe}	EHE Thermal Delivery - Rapid change	CHP Test 1	0.82	10.63	2.08
Q_{ehe}	EHE Thermal Delivery - Slow change	CHP Test 2	0.97	33.08	2.51
Q_{hp}	Heat Pump Thermal Delivery	CHPHP Test 2	0.82	11.47	2.52
Q_{th}	Total Plant Thermal Delivery	CHPHP Test 2	0.55	5.80	2.52
Q_{hp}	Adjusted Heat Pump Delivery	CHPHP Test 2	0.99	56.99	2.52
Q_{th}	Adjusted Total Thermal Delivery	CHPHP Test 2	0.85	12.37	2.52

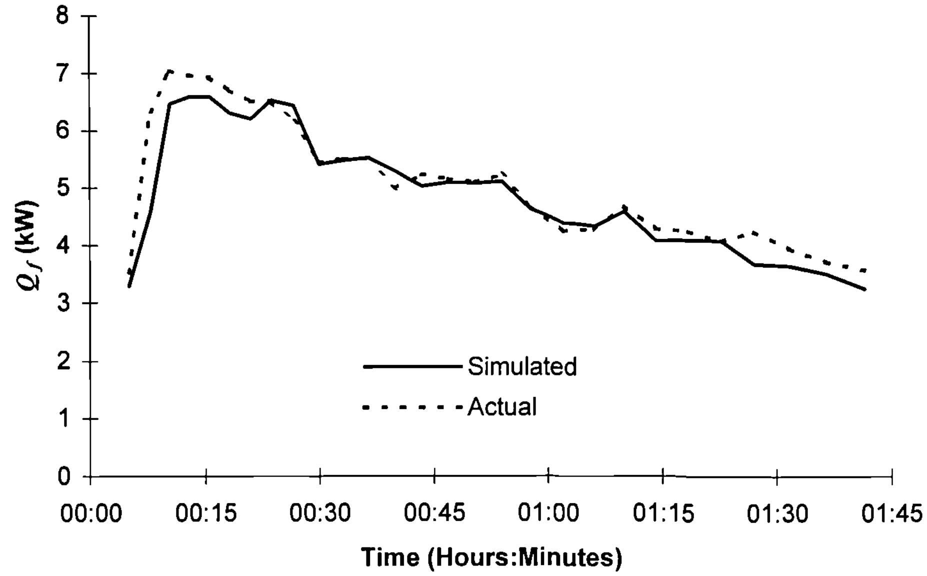


Figure 7.15 - Actual and Simulated Comparison for Fuel Consumption (based on CHP Test 1)

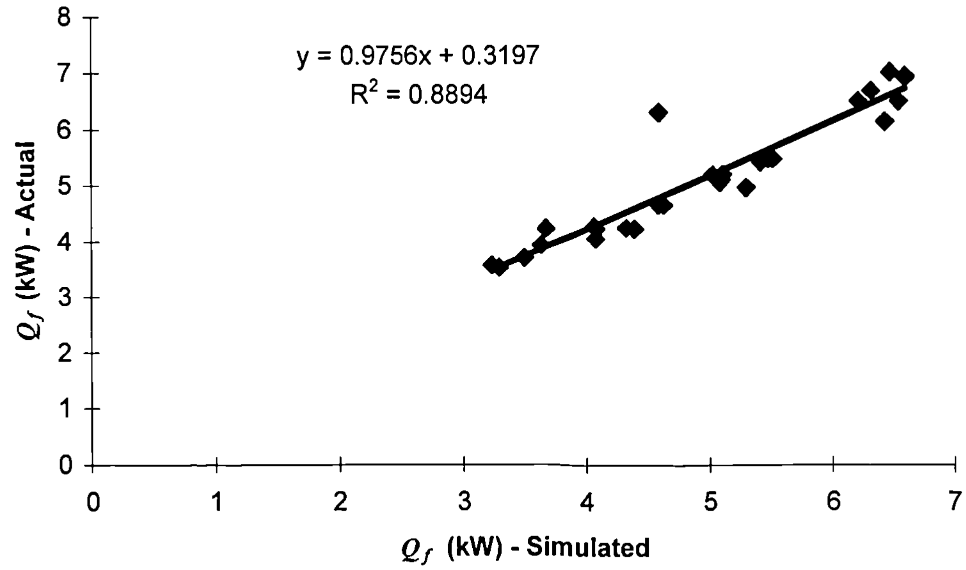


Figure 7.16 - Actual and Simulated Correlation for Fuel Consumption (based on CHP Test 1)

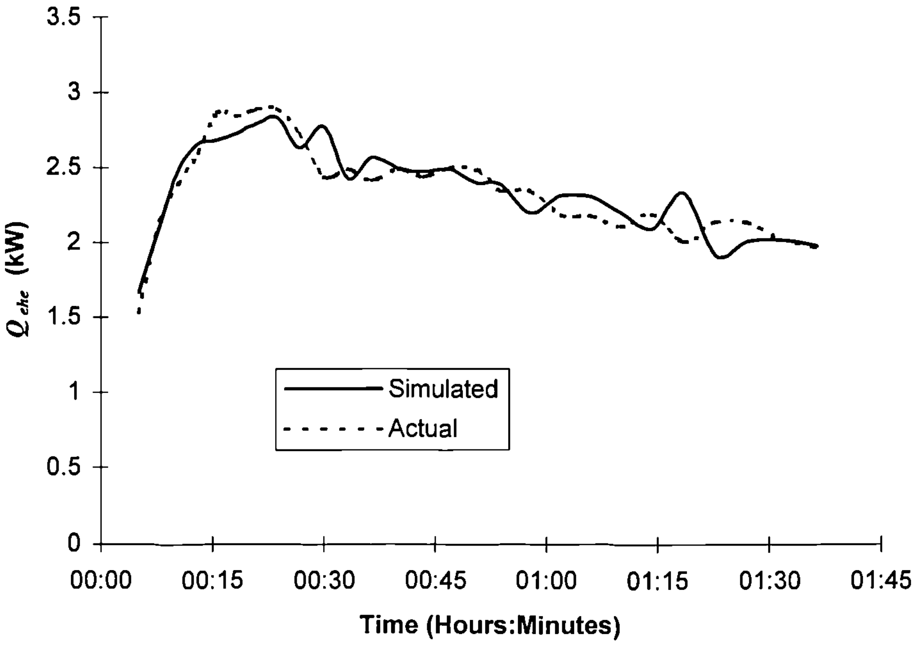


Figure 7.17 - Actual and Simulated Comparison for EHE Thermal Delivery , Q_{ehe} (based on CHP Test 1)

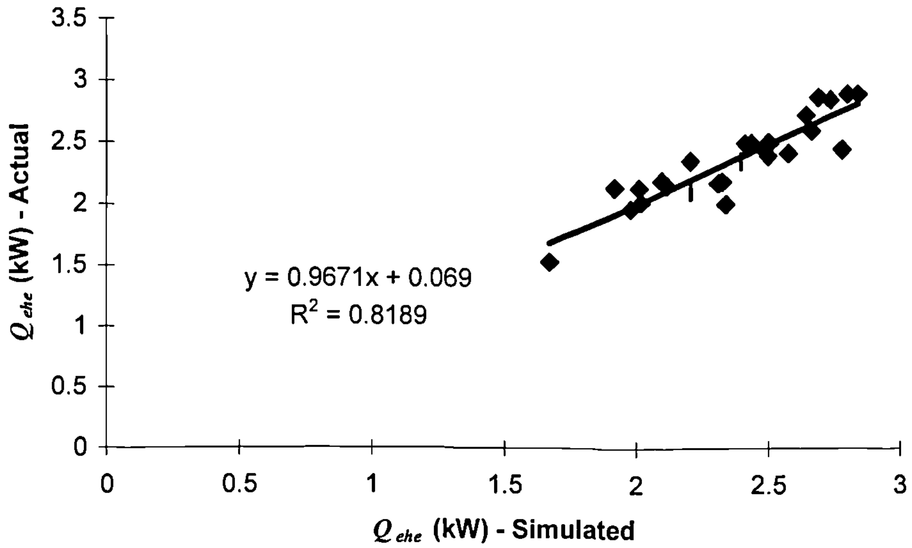


Figure 7.18 - Actual and Simulated Correlation for EHE Thermal Delivery, Q_{ehe} (based on CHP Test 1)

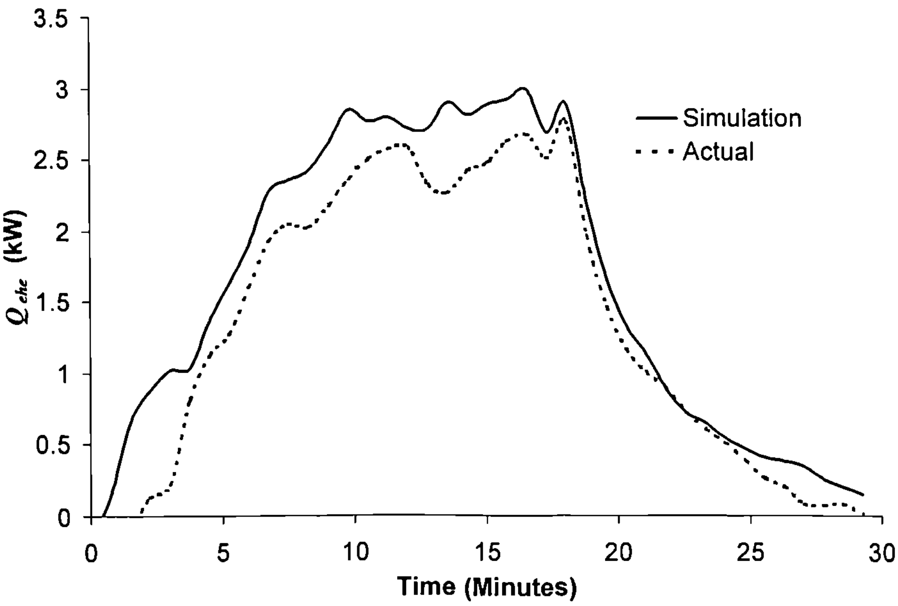


Figure 7.19 – Actual and Simulated Comparison for EHE Thermal Delivery, Q_{ehe} (based on CHP Test 2)

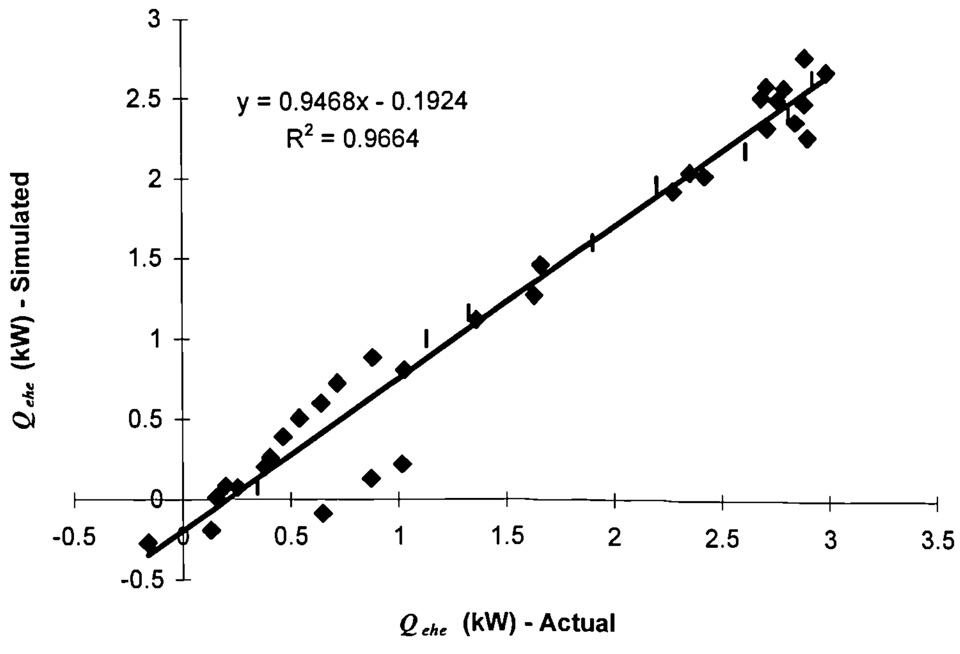


Figure 7.20 - Actual and Simulated Correlation for EHE Thermal Delivery, Q_{ehe} (based on CHP Test 1)

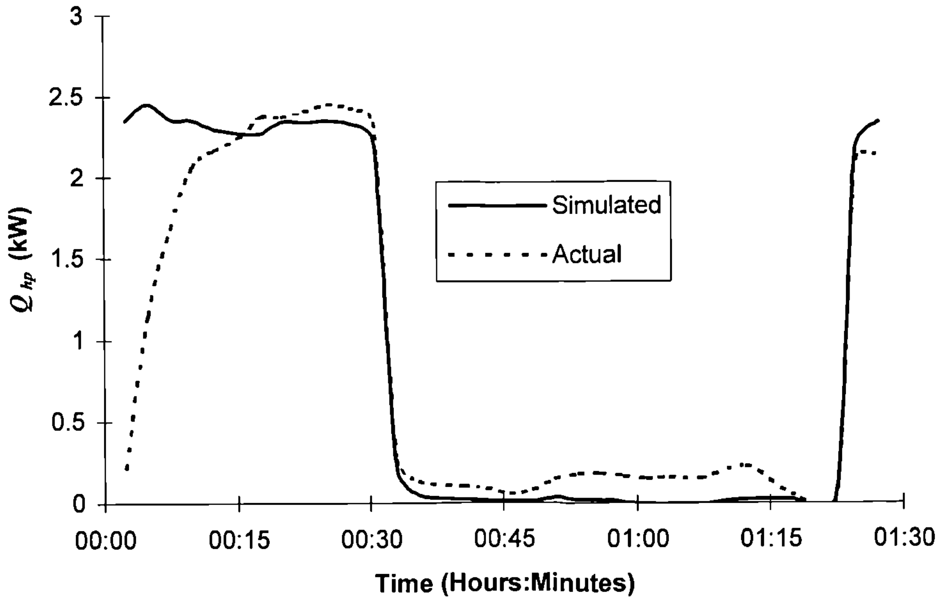


Figure 7.21- Actual and Simulated Comparison for Heat Pump Thermal Delivery, Q_{hp} (based on CHPHP Test 2)

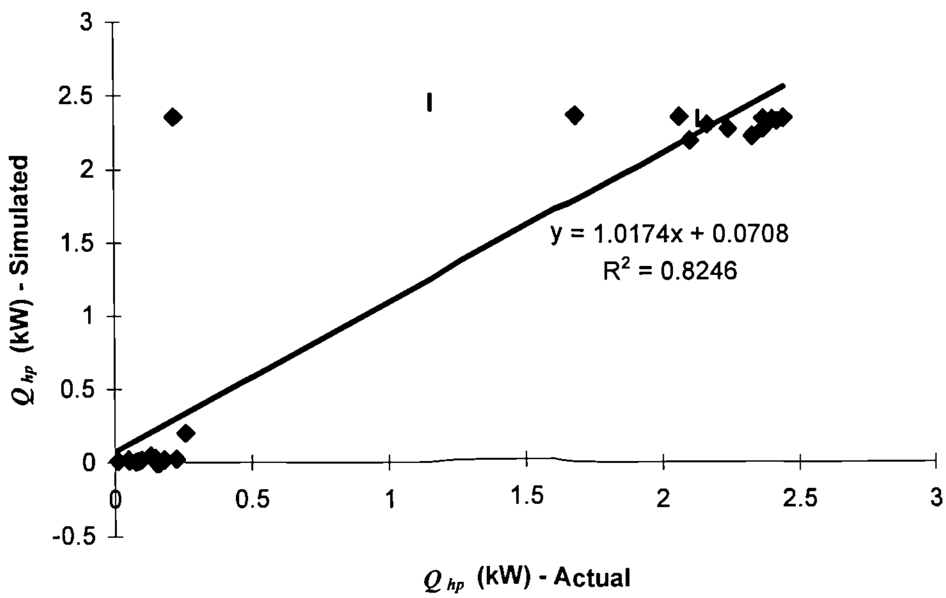


Figure 7.22 Actual and Simulated Correlation for Heat Pump Thermal Delivery, Q_{hp} (based on CHPHP Test 2)

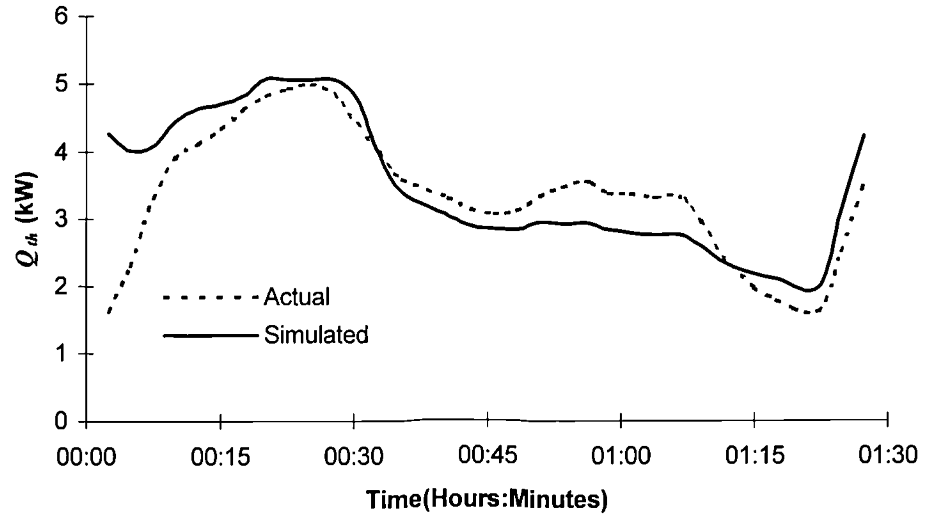


Figure 7.23 - Actual and Simulated Comparison for Total Plant Thermal Delivery, Q_{th} (based on CHPHP Test 2)

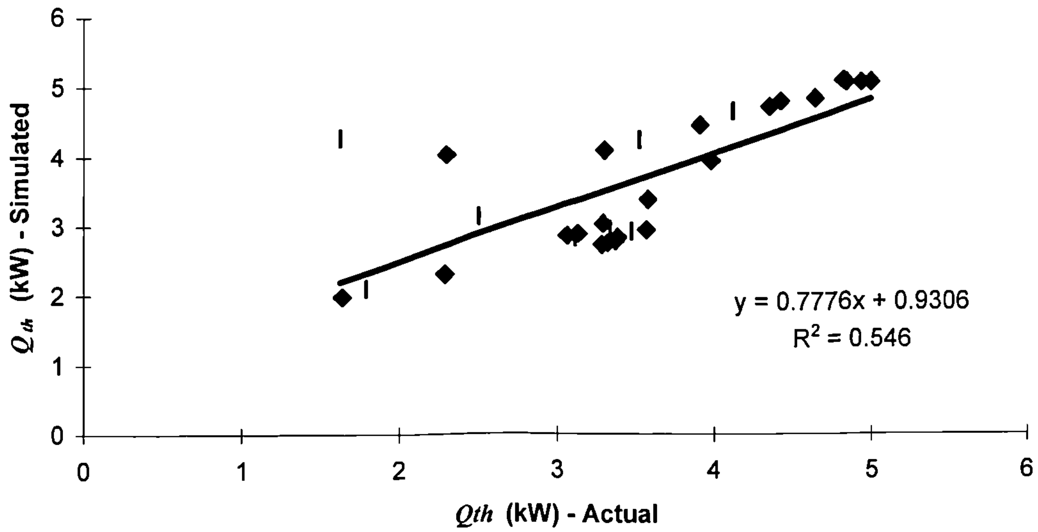


Figure 7.24 - Actual and Simulated Correlation for Total Plant Thermal Delivery, Q_{th} (based on CHPHP Test 2)

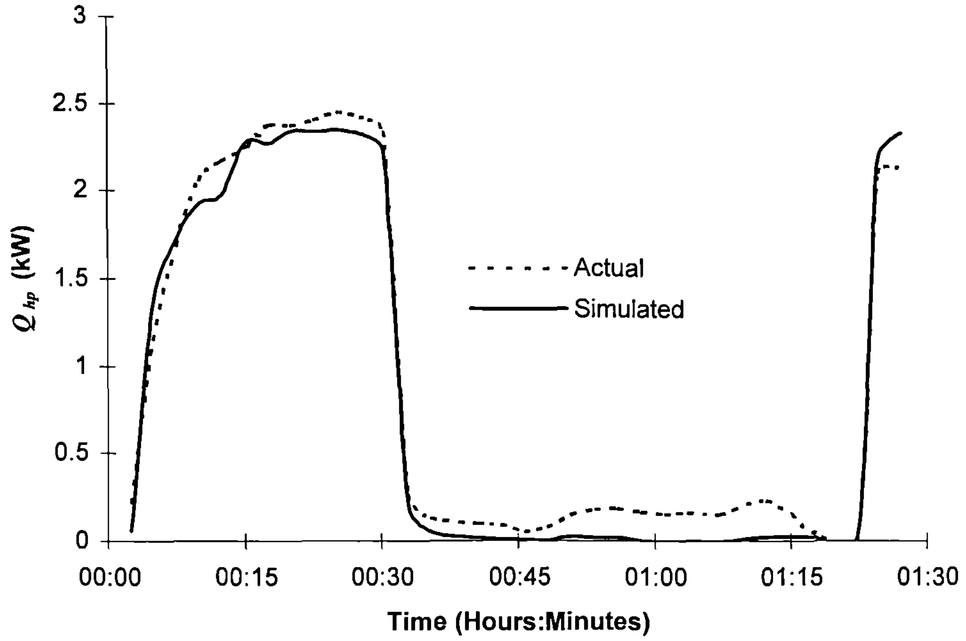


Figure 7.25 - Actual and Adjusted Simulated Comparison for Heat Pump Thermal Delivery, Q_{hp} (based on CHPHP Test 2)

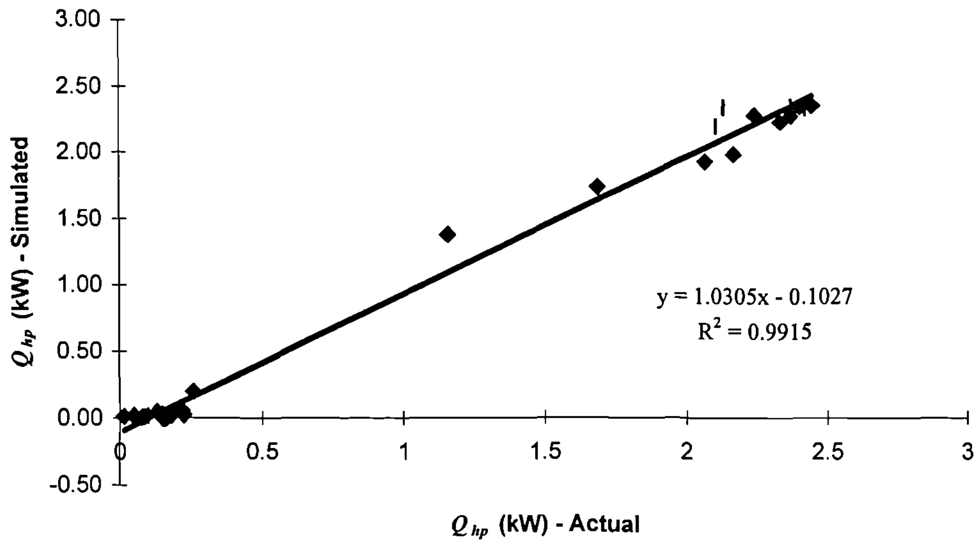


Figure 7.26 - Actual and Adjusted Simulated Correlation for Heat Pump Thermal Delivery, Q_{hp} (based on CHPHP Test 2)

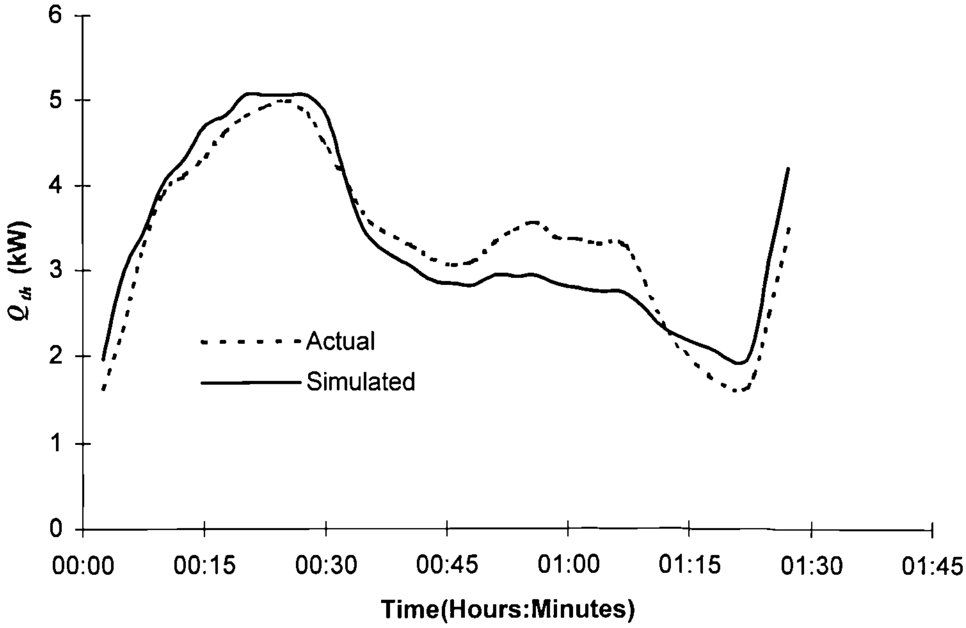


Figure 7.27- Actual and Adjusted Simulated Comparison for Total Plant Thermal Delivery, Q_{th} (based on CHPHP Test 2)

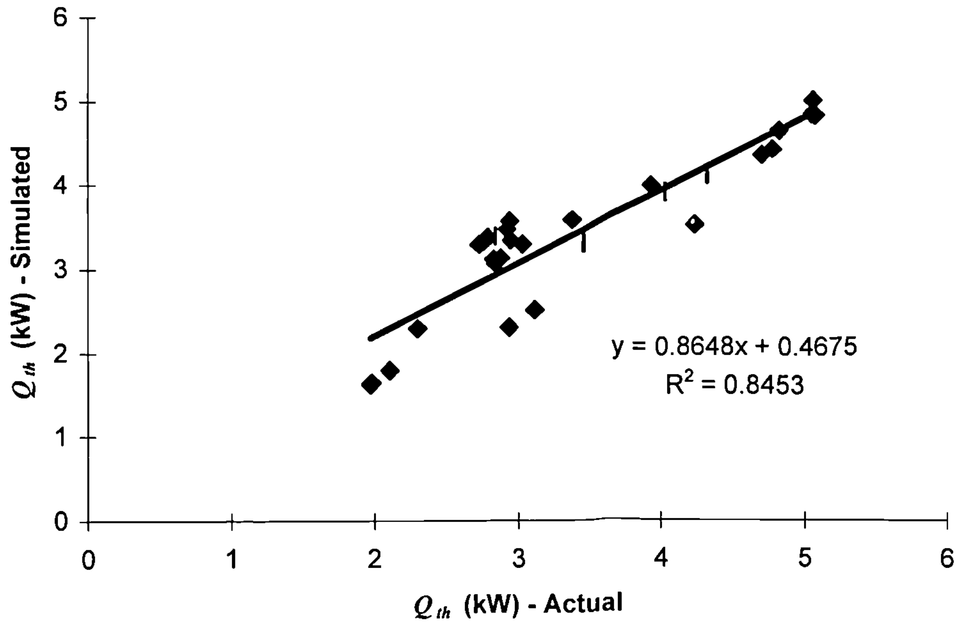


Figure 7.28 - Actual and Adjusted Simulated Correlation for Total Plant Thermal Delivery, Q_{th} (based on CHPHP Test 2)

8. Evaluation of Domestic CHP and CHP/HP

8.1 Introduction

This chapter will utilise experimental results (Chapter 6) and the previously developed concept evaluation model (Chapter 7) to assess both domestic scale CHP/HP and CHP. This analysis will be subsequently used to develop an operational strategy for domestic scale CHP/HP.

Initially, the operation of domestic scale CHP will be examined under varying economic conditions, using actual domestic demand profiles for a 24 hour period, as used in Chapter 4 [39]. This will establish an envelope of economic operation for domestic CHP. Sensitivity to economic parameters will also be established. CHP operation will be used as a base line for further analysis.

Subsequently, CHP/HP operation will be similarly examined and compared to that of CHP. This analysis will indicate relative strengths and weaknesses in the CHP/HP concept. Improvements indicated by the comparative analysis will be modelled, and again compared to the base line CHP case, assessing their viability.

The assessment of both modes of operation will be primarily economic with some supporting environmental analysis. Economic results will be presented as economic savings per day, which will not consider capital costs. The environmental analysis can only be considered to be an approximate indicator, owing to the uncertainties in the calculation of CO_2 emissions. Environmental results will be presented as the percentage CO_2 reduction for a plant configuration compared to conventional systems.

8.2 Assessment of CHP Performance

The following section will assess the performance of domestic scale CHP operation. The validated *Concept Evaluation Model* (see Chapter 7) was configured to simulate the operation of the prototype plant (operating in CHP mode) over a 24 hour period (using Lindford House 18 data). The simulation was repeated to assess CHP operation under varying economic conditions.

The following sections will examine domestic CHP operation with respect to:

- Unit maintenance cost of CHP plant, expressed conventionally as cost per unit electricity generated.
- Unit electricity cost of the utility supply.
- Unit fuel (natural gas) cost of the utility supply.

Additionally, the best case of CHP operation will be established, where the economic conditions are so favourable that the plant operates solely on demand requirements. Finally, an envelope of economic operation for all three economic parameters relative to each other will be established for CHP operation.

Unless stated otherwise, unit electricity price of the utility supplies will be assumed to be £0.0635/kWh[1] and the unit fuel price will be assumed to £0.0135/kWh[2].

[1] Average UK unit price at time of writing [source: The Electricity Association]

[2] Current British Gas unit price at time of writing, Source - BG Customer Services.

8.2.1 CHP Maintenance Unit Cost Effects on Plant Performance

Current unit fuel and electricity prices are readily found from commercial sources, however, maintenance unit costs are subjective. Owing to the experimental nature of the prototype, any value for unit maintenance cost would not be representative of mass produced units. Similarly, applying the unit maintenance cost factors of a large scale CHP installation would also be subjective, as earlier chapters have concluded that conventional CHP technology cannot be applied to domestic installations. The approach taken was to fix fuel and electricity unit prices and vary maintenance cost. The plant simulation was repeated (using the same demand data) for a unit maintenance cost of £0.00/kWh to £0.1/kWh at intervals of £0.001/kWh.

Figure 8.1 compares the thermal demand and the CHP thermal delivery, for a selection of maintenance unit costs (c_m). Similarly, Figure 8.2 compares electrical demand and CHP delivery.

The results for a maintenance free plant ($c_m = 0$) represent the maximum possible plant utilisation for that demand profile. The plant runs when concurrent thermal and electrical demands are present. The plant is modulated primarily on electrical demand, except at point 'A' when the thermal demand reduces and the plant is then modulated appropriately. Demand satisfaction and corresponding load factors are presented in Table 8.1.

When the maintenance unit cost (c_m) is increased to the relatively high level of £0.027kWh, the operation of the plant when satisfying low electrical demands becomes uneconomic. Under such conditions the plant is not run. This can be observed in Figure 8.2, between points A and B. The plant can only be run when the load factor is above 55%, reducing demand satisfaction and operational savings (see Table 8.1).

Further increasing maintenance unit costs (to c_m = £0.0032kWh), again increases the minimum load factor that the plant may efficiently operate at, to 80%. With reference to Figures 8.1 and 8.2, the effect of increasing maintenance costs is evident, with plant operation only taking place during the period of highest demand. Even during this period, the operational savings of the plant would be nominal, with plant running costs being nearly equal to conventional energy costs. Increasing maintenance costs above £0.0033kWh would render the plant uneconomic throughout the 24 hour period.

Table 8.1 Maintenance Unit Cost Effects on CHP Performance

Maintenance Unit Cost £/kWh	**0.000**	**0.027**	**0.032**	**0.033**
Load Factor %	44.24	33.54	08.33	0.00
Electrical Demand Satisfied %	77.78	58.97	14.65	0.00
Thermal Demand Satisfied %	53.95	36.44	07.30	0.00
Operational Savings	0.303	0.022	00.00	0.00
Environmental Savings	9.9	0	0	0

From these results it can be concluded that under present market conditions the maintenance costs of a domestic scale CHP plant would have to be less than £0.0032kWh. This figure represents the absolute maximum value for maintenance costs. The sensitivity of plant operation to maintenance costs must also be examined.

Figure 8.3 compares the CHP plant thermal and electrical outputs (expressed as percentage demand satisfied) totalised for each simulated period against the corresponding maintenance unit costs. Additionally, Figure 8.3 also compares average load factor (for each simulation) with the corresponding unit maintenance cost. In the case of CHP, thermal/electrical demand satisfaction is a function of load factor. Increased maintenance cost has no effect on load factor (and hence demand satisfaction) until a value of £0.023/kWh is reached. At this point the economic viability of plant operation decreases rapidly. Operational performance is extremely sensitive to changes in maintenance costs above £0.027/kWh. Hence a critical value exists (at c_m = £0.027kWh) where the sensitivity to economic factors is so great as to make CHP operation non-viable.

The relationship between load factor and maintenance unit cost is dependant on a number of factors, primarily:

- The part load efficiency of the prime mover. A prime mover with a high part load efficiency will be less sensitive to economic variation, below the critical value. In this case the prime mover has a poor part load efficiency profile (see Figure 6.3).
- The demand profile. The degree of variation in the demand profile will affect the economic sensitivity. A constant demand profile would lead to high sensitivity, as there will be no variation in economic performance for a set of economic conditions. Hence, when the maintenance unit cost approaches the critical value, the lack of variability in the demand profile will result in a marked attenuation of cost effective operation. A highly variable demand profile (as used in these simulations) results in a wide variation in economic performance and reduces sensitivity to maintenance costs near the critical value.

By examining the relationship between operational savings and maintenance costs (see Figure 8.4), a constant decrease in economic viability is evident with an increase in maintenance costs (as would be expected). The relationship between CO_2 emissions reduction and maintenance unit cost, also described by Figure 8.4, is similar to that for load factor (as emissions reduction is a function of load factor). By contrasting the two results, it can be shown that while the economic viability of domestic CHP is marginal at high maintenance unit costs, the environmental advantages are still maintained. From inspection, the environmental advantages remain insensitive until maintenance costs exceed £0.016/kWh, with a corresponding financial saving of £0.101/day.

Although a maintenance unit cost below £0.027/kWh would allow for economic plant operation, a value below £0.019/kWh would maximise economic and environmental advantage. In view of this result, an arbitrary target maintenance cost of £0.015kWh will be used in the following sections for comparison with CHP/HP results (unless otherwise stated).

8.2.2 Electricity Unit Cost Effects on CHP Plant Performance

Figure 8.5 describes the relationship between electricity unit price and operational performance for CHP, using the arbitrary unit maintenance cost of £0.015kWh. It is evident that an increase in electricity unit cost increases plant utilisation and therefore economic and environmental performance (as described by Figure 8.6). A high electricity unit cost will increase the cost effectiveness of CHP generated electricity and hence the plant will be able to operate economically at reduced load factors.

A change in electricity unit cost gives the opposite effect to a change in unit maintenance cost, as can be seen by comparing Figures 8.3 and 8.5, as:

$$savings = \left(w_e c_e + \frac{Q_{th} c_f}{\eta_{boiler}} \right) - \left(w_e c_m + \frac{w_e c_f}{\eta_e} \right) \qquad 8.1$$

Where the first bracketed term is the saving made by the use of CHP generated energy over conventional supplies and the second bracketed term is the costs incurred for plant operation. As the electricity unit cost (c_e) increases, so do the financial savings. Conversely, an increase in unit maintenance cost (c_m) reduces the cost effectiveness of the CHP plant. As both unit costs (c_e and c_m) are multiplied by the electrical output of the plant (w_e), variation in these unit costs have opposing effects (providing fuel unit price remains constant). Additionally, the sensitivity of CHP plant cost effectiveness to a change in electrical unit cost is similar for unit maintenance costs.

8.2.3 Fuel Unit Cost Effects on CHP Plant Performance

Figures 8.7 and 8.8 describe the effect of fuel unit cost variation on operational plant performance and economic/environmental performance, respectively. It can be seen that cost effective plant performance can only be maintained within a relatively small range compared to maintenance costs. By examining equation 8.1, it can be seen that fuel unit cost (c_f) is present in both the savings and cost terms i.e.:

Saving component: $\dfrac{Q_{th} c_f}{\eta_{boiler}}$

and cost component: $\dfrac{w_e c_f}{\eta_e}$

When conditions are such as to make the above components equal i.e.:

$$\frac{Q_{th} c_f}{\eta_{boiler}} = \frac{w_e c_f}{\eta_e}$$

a change in fuel unit cost (c_f) has no effect. However, when the saving component has a greater value than the cost component i.e.:

$$\frac{Q_{th} c_f}{\eta_{boiler}} > \frac{w_e c_f}{\eta_e}$$

a higher fuel unit cost (c_f) will increase plant cost effectiveness and hence favour CHP generated energy. Conversely, when the cost component is greater than the saving component, the CHP plant cost effectiveness is reduced.

Both plant electrical efficiency (η_e) and plant thermal output (Q_{th}) are functions of plant mechanical output (w_e), hence the ratio of saving component to the cost component will be dependant on w_e. This ratio will have an effect on the cost effectiveness of plant operation.

Let β be the ratio of the savings component to the cost component, i.e.:

$$\beta = \frac{Q_{th}}{\eta_{boiler}} \Big/ \frac{w_e}{\eta_e} \qquad (8.2)$$

If $\beta = 1$, then fuel cost variation has no effect on plant output (as stated previously). If $\beta > 1$, plant cost effectiveness is increased with increased fuel costs. If $\beta < 1$, then plant cost effectiveness is reduced with an increased fuel unit cost. Figure 8.9 relates the values for β to the corresponding electrical plant outputs (w_e), where Q_{th} and η_e are calculated from expressions derived in Sections 6.2.2 and 6.2.1 respectively.

With respect to Figure 8.9, the line A-A' (at an engine load of 0.14kWe) represents the plant conditions that apply when the fuel unit cost is increased to the point at which CHP operation becomes non-viable. Line B-B' (at an engine load of 0.48kWe) represents the conditions that apply when the fuel unit cost is reduced to the point where operational plant performance is at a maximum.

Although the β / w_e relationship is intended to be used for future comparison with CHP/HP operation, some intuitive results can be confirmed:

- Poor boiler efficiency (η_{boiler}) favours CHP operation, as in this case when $\eta_{boiler} < 45\%$, $\beta > 1$.
- A good plant heat to power ratio improves CHP cost effectiveness: as Q_{th} increases, then β increases.
- Poor engine full and part load efficiencies reduce the value of β, and hence cost effectiveness, as expected.

8.2.4 Envelope of Economic Operation

Sections 8.2.1 to 8.2.3 have examined the operational and economic performance of CHP operation with respect to one variable unit cost. Figure 8.10 describes the envelope of economic operation for all economic conditions with respect to each other.

The simulation was run for the same test period for all possible combinations of the three unit costs. Figure 8.10 illustrates the relationship between the maximum fuel unit cost that will allow for economic operation at a given electricity unit cost and selected maintenance unit costs. The area below each line represents an envelope of economic operation for that maintenance cost. The magnitude of plant cost effectiveness is not considered by Figure 8.10.

The results summarised in Figure 8.10 indicate the economic conditions necessary to allow domestic scale CHP to be economically viable. The importance of a low maintenance cost is again reinforced, as small changes will significantly reduce economic viability. These results will be used later in this chapter as a comparison to CHP/HP simulations.

8.3 Assessment of CHP/HP Performance

The following section will examine the simulated performance of domestic CHP/HP. Initially, the prototype configuration will be simulated using current unit costs and the arbitrary maintenance costs derived in Section 8.2.1. The effect of heat pump characteristics will be analysed prior to the establishment of unit price sensitivities and an envelope of economic operation (as established for CHP operation in Section 8.2). This analysis will be subsequently used to compare CHP and CHP/HP operation.

8.3.1 Simulation of Prototype Plant Configuration

The prototype plant required an electrical surplus above 0.87kW (see Section 6.2.3) to allow CHP/HP operation. The demand profile utilised in the simulation was such that, given the engine size, a sufficiently high surplus could not be realised. Hence plant operation was identical to CHP operation (under equivalent economic conditions).

8.3.2 The Effect of Heat Pump Part Load COP

As demonstrated in Sections 6.3 and 6.4, under steady state high demand conditions, CHP/HP has clear thermodynamic advantages over conventional CHP. However, when prototype plant configuration is not suited to the highly variable demand environment that would be experienced in a domestic application, it will be necessary to modulate the heat pump. This section will examine the effect of part load heat pump COP on CHP/HP operation by:

- Establishing a number of part load COP characteristics.
- Describing demand and delivery profiles for different part load COP characteristics.
- Analysing the operational, financial and environmental performance for a range of part load COP characteristics.

As detailed in Section 7.4.6.3, provision was made in the concept evaluation model to vary heat pump part load characteristics by applying an exponential function, i.e.:

$$COP = \left(1 - e^{w_s j_{COP}}\right) COP_{\max} + COP_0$$

Where COP_{max} is the maximum heat pump COP achievable in steady state experimental conditions and j_{COP} is defined as the exponential constant. The value of j_{COP} defines the heat pump part load characteristic. In the following analysis, j_{COP} will be quoted when defining a part load characteristic. Figure 8.11 describes some generated COP part load characteristics that will be referred to in the following sections.

8.3.2.1 Demand and Delivery Profiles for Varying COP

Figure 8.12 describes the thermal demand for the test case (as in Section 8.2) and the simulated thermal plant responses for three different part load COP characteristics. At this stage, little variation in thermal delivery is observed for the delivery profiles. During the mid-day section of the simulation (11:00 to 16:00 hours) all the delivery profiles match the demand profiles. During this stage, the plant is running in CHP/HP mode. In the evening period (16:00 to 22:00 hours), electrical demand (see Figure 8.13) is higher than the maximum rated engine/generator output, and consequently the plant defaults to CHP operation. This is evident in the reduction in plant thermal output. With reference to Section 4.5.1, the mechanics of such profiles have been previously covered during the preliminary model analysis. The electrical delivery for all part load COP characteristics follows the electrical demand closely, owing to CHP/HP operation. Although the operational performance of different part load characteristics appears to be marginal, further analysis is required.

8.3.2.2 The Effect of j_{COP} on Operational Performance

The simulation was repeated using a wide range of j_{COP} values. The percentage demand satisfactions and load factors were totalised for each simulation, in a similar method to that employed in Section 8.2.

The result of these simulations is illustrated in Figure 8.14. No change in operational performance is evident, where j_{COP} exceeds a value of 8. With reference to Figure 8.11, a j_{COP} value of 10 still exhibits relatively poor part load COP characteristics compared to a higher value, such as 50, which is virtually linear. Such a linear profile could not be obtained practically and will be referred to as an ideal case in further comparisons. As little operational improvement is evident with j_{COP} values greater than 8, this particular profile may be considered to be a tolerable part load characteristic for operational performance.

Figure 8.16 examines the operational performance of low j_{COP} values in greater detail. Analysis of the parameters shows:

- The electrical demand satisfaction for all heat pump characteristics is constant. This is due to CHP/HP operation, which prioritises electrical delivery.
- The plant thermal delivery (the total EHE and heat pump outputs) rises with improved part load COP characteristics. With higher COPs, more thermal energy is delivered for a given electrical surplus.
- Load factor exhibits a maximum when j_{COP} = 4.5. As the j_{COP} rises, CHP/HP becomes more economic and hence the engine load is increased. As j_{COP} is increased further, the thermal delivery per unit heat pump electrical input is increased. Hence less work input is required in meeting thermal demand, resulting in a reduction in load factor.

8.3.2.3 The Effect of j_{COP} on Financial and Environmental Performance

Figure 8.16 describes the relationships between j_{COP} and financial and environmental performance. A high degree of sensitivity is evident for values below 10. Once j_{COP} values exceed 15, there is marginal improvement in financial and environmental performance.

These results also highlight the sensitivity of CHP/HP operation to poor part load COP characteristics. The surplus generating capacity available is always below 0.5kW - the effect of poor part load COP is exacerbated when CHP/HP is operating in this region. The region is described in Figure 8.11 as line A-A'. Hence CHP/HP financial and environmental performance has a high degree of sensitivity to low values of j_{COP}.

An alternative to improving part load COP characteristics would be to use a larger engine/generator set, so that more surplus generating capacity is available. This would potentially increase electric and thermal demand satisfaction and increase the cost effectiveness of CHP/HP operation in supplying high demands.

An ideal part load characteristic is not achievable and would not be appropriate to use in comparative analysis between CHP and CHP/HP performance. Hence two values will be used in further analysis (with reference to Figure 8.16):

- A minimum operational tolerable part load COP characteristic (j_{COP-i}), of 8.
- A minimum financial/environmental tolerable part load COP characteristic (j_{COP-ii}) of 15.

8.4 Comparative Analysis of CHP and CHP/HP Performance

In the following sections, the financial, environmental and operational performance of CHP/HP will be presented and compared to that of CHP. CHP/HP performance will be examined with respect to varying unit maintenance, electricity and fuel costs, in a similar manner to that undertaken for CHP operation in Section 8.2. This comparative analysis will define the economic conditions for appropriate CHP/HP implementation and will develop a control strategy for a CHP/HP. Analysis will use financial and environmental performance as primary indicators and operational performance as supporting data.

8.4.1 Effects of Maintenance Unit Costs on CHP/HP Performance

Figure 8.17 shows the effect of maintenance cost on the financial performance of CHP/HP and CHP operation. Comparing both modes of operation, a lower maintenance cost favours CHP/HP operation. Once the maintenance unit cost exceeds £0.01/kWh, the financial performance of the two modes of operation becomes similar. To identify the mechanism causing this effect, an examination of the effects of maintenance unit costs on load factor (see Figure 8.18) is necessary. When maintenance costs are relatively low (below £0.01/kWh) both modes of operation maintain high load factors. CHP operation is dictated by electrical demand, while CHP/HP operation utilises surplus generating requirement to increase thermal delivery. Owing to the effect of the engine part load efficiency, as maintenance costs increase, then the plant generated electricity becomes less economic for low electrical demand. When the maintenance cost is increased further, then only when the engine is under maximum load, is plant generated electricity cost effective (compared to utility supplied power). Similarly, when maintenance unit costs are increased, then heat pump thermal delivery becomes less economic compared to boiler thermal delivery. Poor part load heat pump COP compounds this effect.

The overall result is that as maintenance costs increase, the cost effectiveness of CHP/HP operation is marginalised and only becomes effective during periods of high electrical demand, when little or no generating surplus is available. Hence the CHP/HP plant tends to run as a CHP plant with little or no heat pump operation. Therefore, when CHP/HP operation is marginalised by increasing maintenance unit costs, the plant behaves as a CHP plant, giving rise to similar financial performance. This point is reinforced by Figure 8.19, which compares CHP and CHP/HP electrical deliveries, which show identical characteristics with respect to maintenance cost.

Figure 8.20 shows the effects of maintenance unit cost in CHP and CHP/HP environmental performance. As environmental performance is a function of load factor, the effect of maintenance cost is similar for both parameters. Until CHP/HP operation is compromised by increasing maintenance unit costs, surplus generating capacity is used to deliver heat via the heat pump - displacing boiler delivered heat and hence reducing carbon dioxide emissions. Hence for CHP/HP, the reduction of carbon dioxide emissions is greater than that for CHP operation, until plant operation is rendered uneconomic by increasing maintenance costs. With reference to Figure 8.20, it can be seen that the environmental performance of CHP/HP operation for $j_{COP\text{-}ii}$ (minimum financial/ environmental part load COP characteristic) is better for high maintenance costs than $j_{COP\text{-}i}$ (minimum operational part load COP characteristic) as the higher part load heat pump COP allows for more economic use of surplus generating capacity.

8.4.2 The Effect of Electricity Unit Costs on CHP/HP Operation

With reference to Figure 8.21, it is evident that an increase in electricity unit costs does not favour CHP/HP or CHP operation. At low electricity unit costs, neither mode of operation can compete with utility supply. When the electricity unit costs become high enough to allow for cost effective operation, CHP/HP operation is marginalised by the same mechanism identified in Section 8.4.1, where a CHP/HP plant will operate as a CHP plant. As electrical unit cost is increased further, the CHP/HP operation becomes marginally more cost effective.

At low electricity unit costs, the additional thermal delivery of CHP/HP operation is small, giving rise to a minimal increase in financial savings. At high electricity unit costs, the additional CHP/HP delivery is significant. However, the contribution that the increased thermal delivery makes to financial savings is low compared to that made by plant generated electricity. As CHP/HP and CHP operations satisfy electrical requirements to approximately the same degree, and given the relatively small financial contribution the additional CHP/HP thermal delivery makes to the financial performance, both types of operation are similar with respect to varying electricity unit cost.

Although the additional thermal delivery of CHP/HP operation has a marginal effect on financial performance, the displacement of boiler emissions has a significant effect on comparative environmental performance (see Figure 8.23). With respect to environmental performance, a higher electricity unit cost favours CHP/HP operation.

8.4.3 The Effects of Varying Fuel Price on CHP/HP Operation

Figure 8.24 compares the daily financial savings of both CHP and CHP/HP operation with respect to varying fuel price. At a low value of fuel unit cost, CHP is more cost effective than CHP/HP. As fuel costs increase, the CHP/HP operation becomes increasingly cost effective relative to CHP operation.

Both modes of operation return high financial savings for low fuel unit costs. When fuel unit costs are low, plant generated electricity is far less expensive than utility supplied electricity, returning a high saving. As unit fuel prices increase, the cost effectiveness of plant generated electricity decreases with respect to utility supply until no saving is returned.

As noted in Section 8.4.2, the financial value of satisfying electricity demand is far higher than that of thermal demand: with a low fuel unit cost, thermal energy has an extremely low financial value. In CHP/HP operation, the additional heat pump thermal delivery (as shown in Figure 8.25) incurs a maintenance cost, due to the additional load on the engine/generator set. At low values of fuel unit cost, the incurred maintenance cost renders the additional heat pump supplied heat uneconomic compared to boiler supplied heat. Hence, for low values of fuel unit cost, the CHP operation is more cost effective than CHP/HP operation, as the additional delivery of heat is supplied by a boiler. As fuel unit costs are increased, the additional heat pump supplied heat becomes more cost effective than boiler supplied heat (described in Figure 8.24).

The envelope of economic CHP/HP operation is extended to higher values of fuel unit costs compared to CHP operation as a consequence of the high cost effectiveness of heat pump supplied heat. This is evident in the electrical delivery of CHP/HP operation compared to that of CHP operation (see Figure 8.26). This extension of operation has additional benefits for environmental performance. As with maintenance and electricity unit costs, the environmental performance of CHP/HP is significantly better than for CHP operation, as demonstrated in Figure 8.27. As with the previous economic cases, this is due to the higher thermal delivery of CHP/HP over CHP operation (see Figure 8.25

8.4.4 Examination of the β Function for CHP/HP Operation

As described in Section 8.2.3, by comparing the cost and savings components of the financial savings function (Equation 8.1), the effect of varying fuel costs on plant economics can be examined. From Equation 8.2:

$$\beta = \frac{Q_{th}}{\eta_{boiler}} \bigg/ \frac{w_e}{\eta_e}$$

The β function requires the calculation of the total plant thermal output (Q_{th}), which in the case of CHP/HP operation is a function of heat pump COP, engine load and surplus generating capacity, i.e.

$$Q_{th} = Q_{ehe} + COP(w_e - D_e) \qquad (8.3)$$

For CHP operation, only engine load (w_e) need be taken into account when calculating β. However, in CHP/HP operation, the electrical demand (D_e) must also be considered.

The β function was calculated for a range of both engine loads and electrical demands, using parameters previously used in the concept evaluation model. The results are presented in Figure 8.28.

With reference to Figure 8.28:

- For values where electrical demand is equal to or greater than engine load (i.e. $D_e \geq w_e$), CHP operation is invoked, as no surplus generating capacity is available for heat pump operation. CHP operation is represented on the Figure by the area ACD.
- The limit of CHP operation is represented by the line AC, which is identical to the function described in Figure 8.9.
- Values for which there is no electrical demand ($D_e = 0$) represent the case of a gas driven heat pump (line AB).

The results for CHP/HP operation β function can be divided into four groups:

- CHP operation (as noted above) - where β values are that of CHP operation, i.e. $\beta < 1$. Hence an increase in fuel cost will reduce the cost effectiveness of plant operation. In these cases, there is a trend for β to decrease with increased engine load (w_e) - as a consequence of the relationship between w_e and engine/generator efficiency (η_e)
- Low heat pump utilisation - where little surplus generating capacity is available for the heat pump. In these cases, β <1, hence a rise in fuel unit cost will reduce plant cost effectiveness. The trend is for β to increase with increased engine load.
- High heat pump utilisation - where a significant generating surplus is available for heat pump use. In these cases, the values for β approach or exceed unity (i.e.: $\beta \geq 1$), where engine load is high. Hence for high engine loads, an increase in fuel unit cost will increase plant cost effectiveness.
- Gas driven heat pump operation - where all the generating capacity is available for heat pump use. In all but the lowest engine loads, this type of operation returns $\beta > 1$, hence increasing plant cost effectiveness with increased fuel cost.

8.4.5 Envelope of Economic Operation for CHP/HP

The envelopes of economic operation for CHP/HP, with respect to varying unit maintenance costs, is presented in Figure 8.29. Figure 8.30 compares the envelopes of economic operation for both CHP and CHP/HP at a relatively low maintenance cost of £0.01/kWh. The envelope of economic operation is significantly greater for CHP/HP operation than for that of CHP.

For low electricity unit costs the maximum unit fuel cost that will allow for economic operation is similar for both modes of operation. At low electricity unit costs, the CHP/HP plant will tend to act as a CHP plant due to the mechanism described in Section 8.4.1 and referred to in Section 8.4.2.

With an increase in electricity unit costs, CHP and CHP/HP operation becomes increasingly cost effective (see Section 8.4.2). However, the incorporation of the heat pump allows for CHP/HP operation with a higher fuel unit cost compared to that of CHP – highlighting the advantages of heat pump incorporation. As stated in Section 8.2.4, the envelope of economic operation does not report the magnitude of cost effectiveness. Although the envelope of CHP/HP economic operation may be greater than that for CHP operation, Sections 8.4.2 and 8.4.3 indicate that actual financial savings will similar for both modes. However, extending the envelope of economic operation will increase the environmental effectiveness of domestic CHP: this point will be discussed in Section 8.5.3.

8.5 Summary and Discussion of CHP and CHP/HP Operation

In the previous sections, the prototype plant's domestic operation (for CHP and CHP/HP modes) has been simulated using representative demand data under different economic conditions. This section will summarise the findings of these simulations and discuss aspects of the analysis that will also have a bearing on the implementation of domestic CHP/HP, while general conclusions will be made in Chapter 9.

8.5.1 Economic Feasibility of Domestic CHP and CHP/HP

The analysis of CHP and CHP/HP simulated operation has demonstrated that under current economic conditions both modes of operation are economically viable, provided that maintenance unit costs are below £0.027/kWh. However, to realise a reasonable level of performance, the maximum maintenance unit cost is £0.019/kWh (for both modes of operation). Therefore the target maintenance cost was set at £0.015/kWh.

Under this assumption, the maximum fuel unit cost that will allow for cost effective CHP/HP operation was £0.02/kWh, compared to £0.018/kWh for CHP operation. The minimum electricity unit price that would allow for economic operation for both modes is £0.05/kWh.

8.5.2 Comparison of Financial Performance

The magnitude of financial savings returned by CHP and CHP/HP operation depends on relative values of unit costs. It has been demonstrated in Section 8.4.1 that CHP/HP performs better than CHP with a maintenance unit cost below £0.01/kWh. There is no significant variation in financial performance with respect to varying electricity unit costs between the two modes of operation (see Section 8.4.2). A low fuel unit cost favours CHP over CHP/HP, while a high fuel unit cost favours CHP/HP (see Section 8.4.3).

8.5.3 Comparison of Environmental Performance

CHP/HP has significant environmental benefits over CHP in a domestic installation. The previous sections have consistently demonstrated that CHP/HP operation further reduces carbon dioxide emissions by 30 to 40% over CHP operation for identical economic conditions. Heat pump incorporation maximises the environmental performance within the envelope of economic operation, as demonstrated by Figures 8.23 and 8.27. The discrepancy between environmental and financial performance is due to the relatively low cost of utility supplied gas, as noted in Sections 8.4.2 and 8.4.3.

8.5.4 Effects of Heat Pump Part Load COP

It has been demonstrated that CHP/HP performance is marginalised by poor part load heat pump COP. The values for β (see Section 8.4.4) show that when the generating surplus is low, heat pump incorporation has a minimal effect on plant performance. However, when a high generating surplus is available, heat pump incorporation makes a significant difference to plant performance and reduces the plant sensitivity to fuel cost variations. This will be taken into account when establishing an operational strategy for a domestic scale CHP/HP plant. Section 8.3.2 demonstrates the criticality of part load heat pump COP on plant performance.

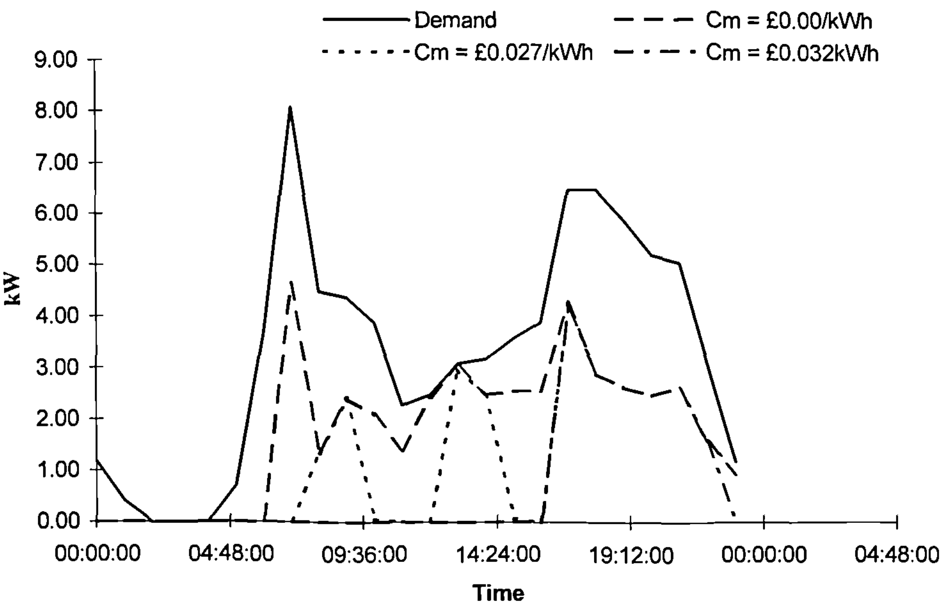


Figure 8.1Thermal Demand and CHP Supplies

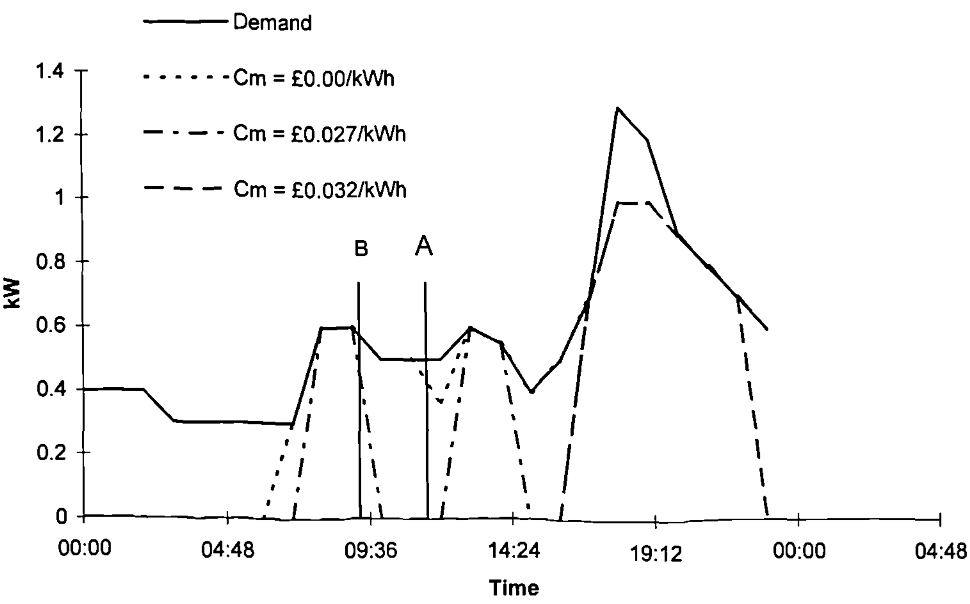


Figure 8.2 Electrical Demand and CHP Deliveries

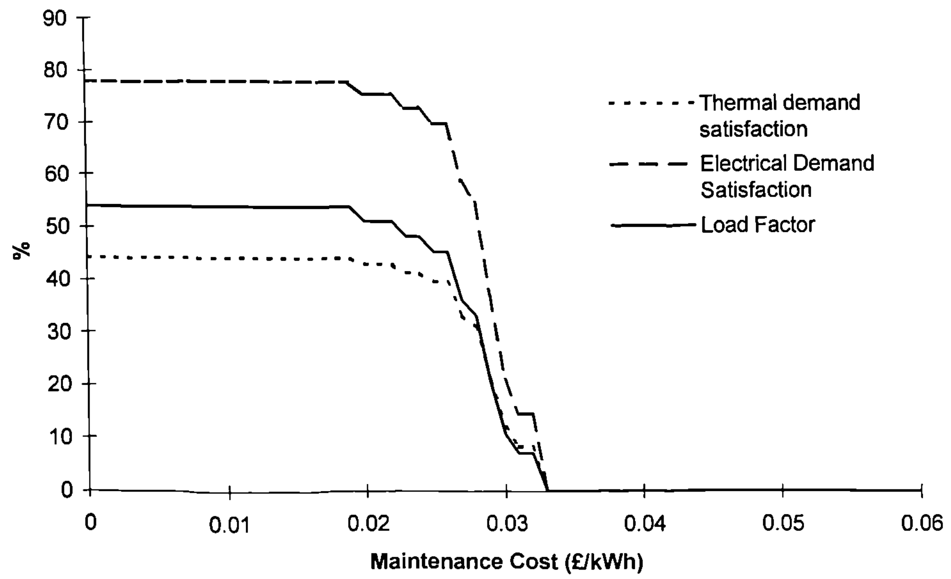


Figure 8.3 CHP Operational Parameters - Varying Maintenance Cost

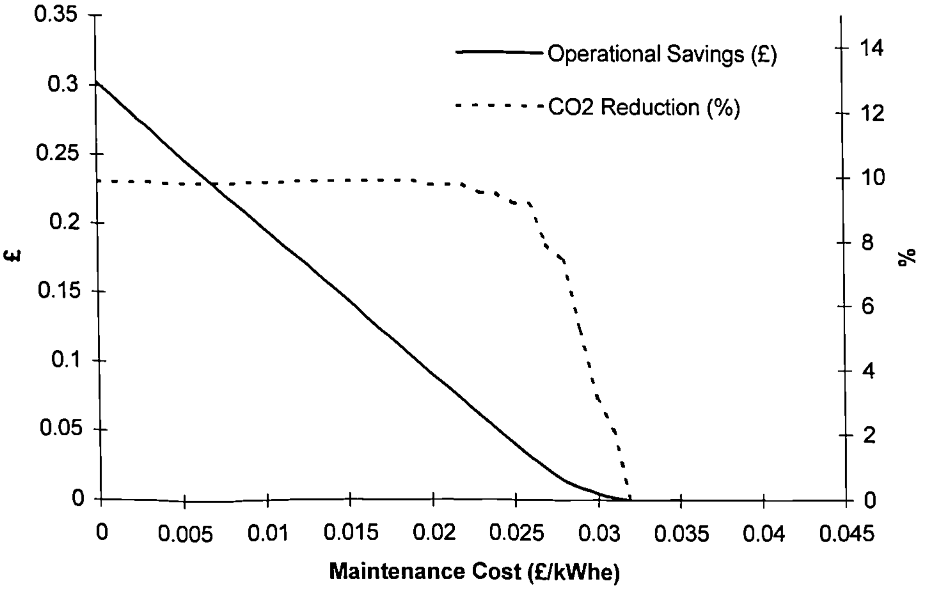


Figure 8.4 Operational Savings and Carbon Dioxide Emissions Reduction

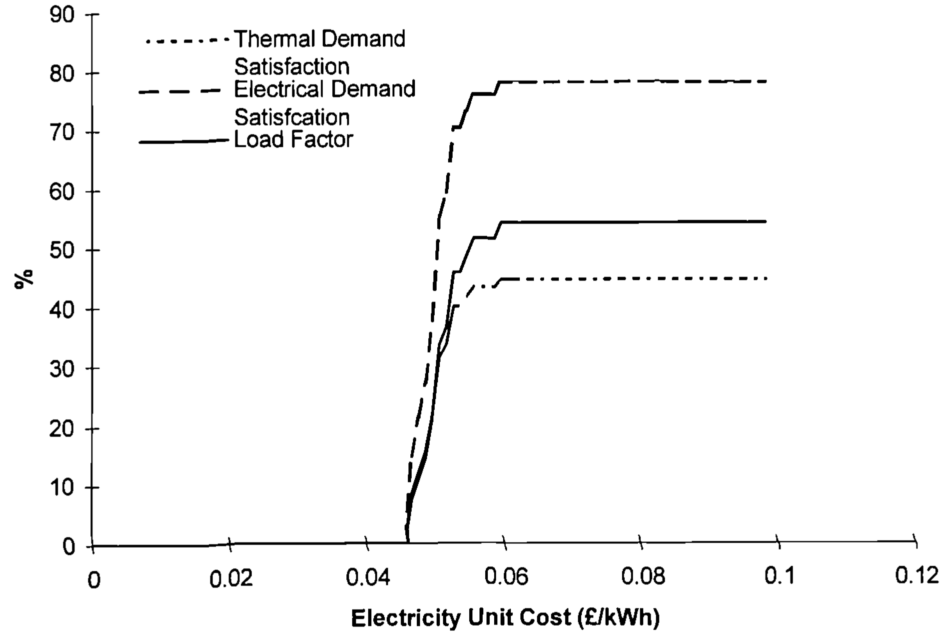


Figure 8.5 Operational Parameters - Varying Electricity Unit Cost

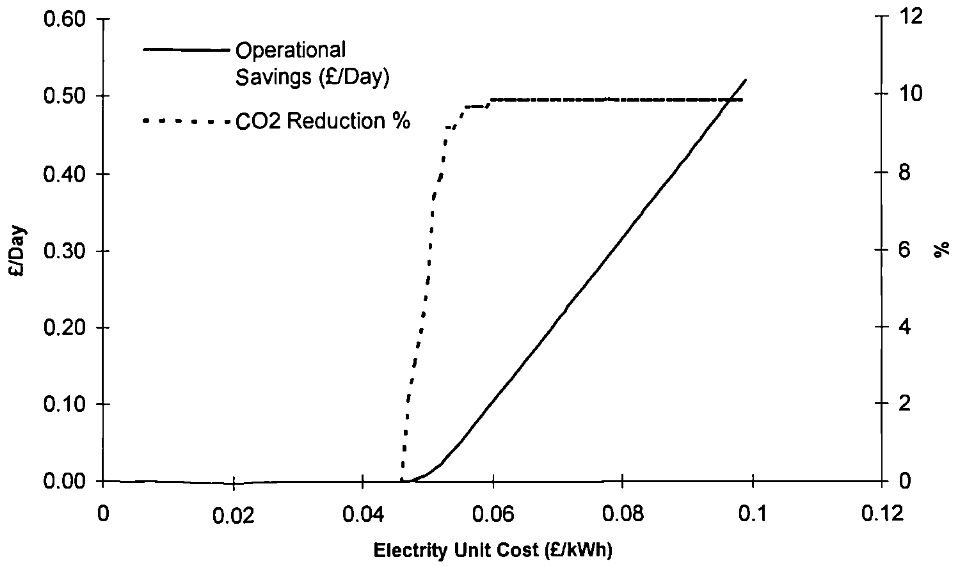
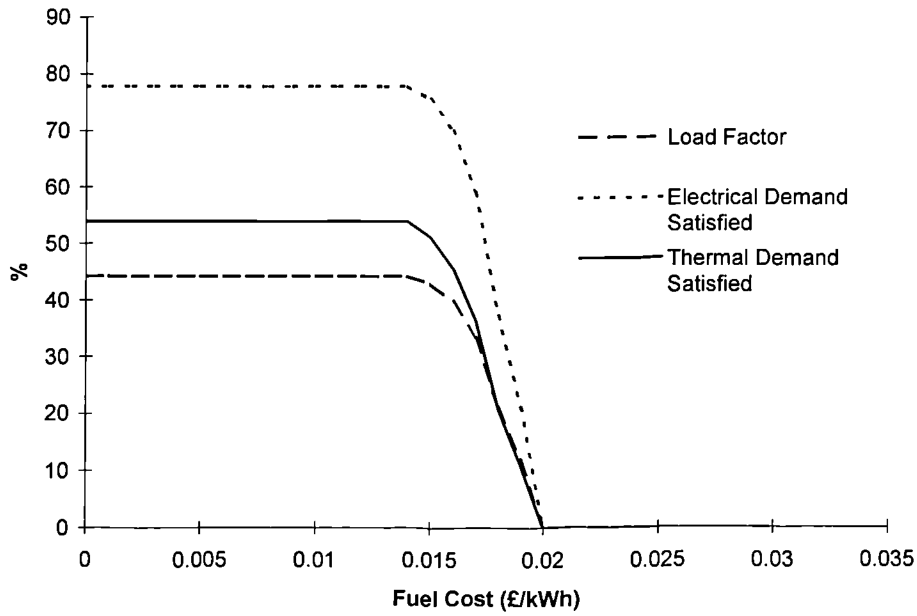


Figure 8.6 Operational Savings and Carbon Dioxide Emissions Reduction vs Electricity Unit Price



Graph 8.7 Operational Parameters vs Varying Fuel Cost

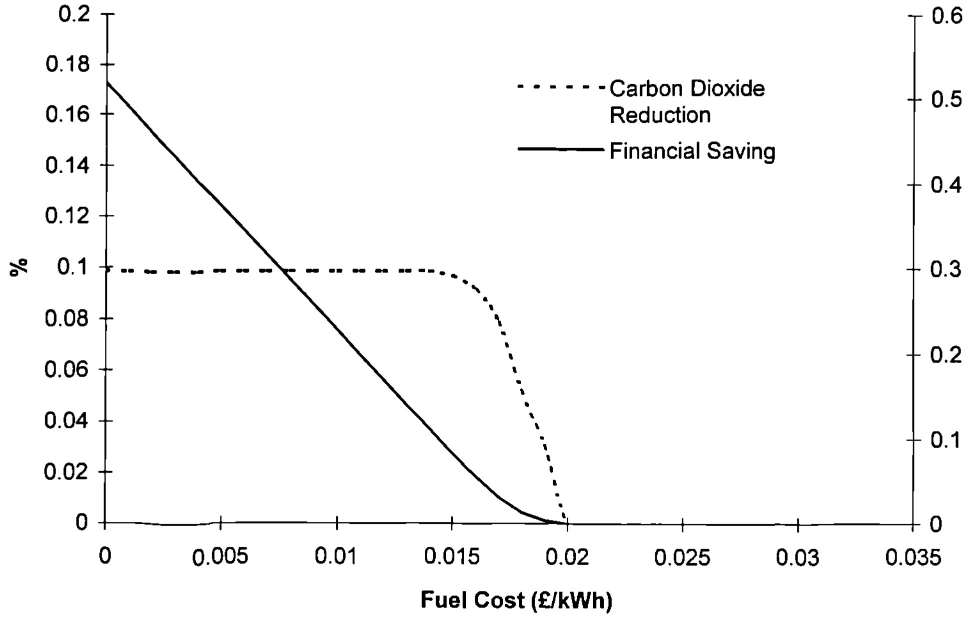


Figure 8.8 Financial and Environmental Savings vs Unit Fuel Cost

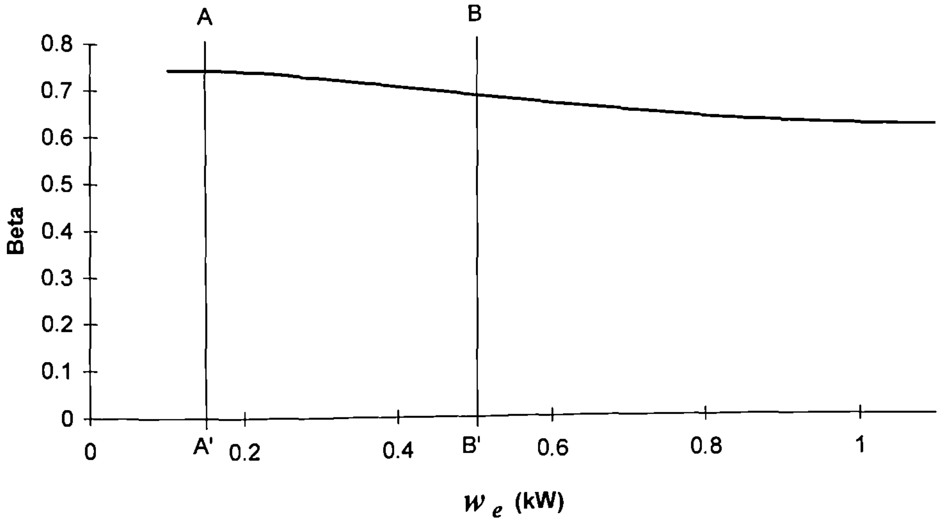


Figure 8.9 Beta Ratio vs Engine Load for CHP Operation

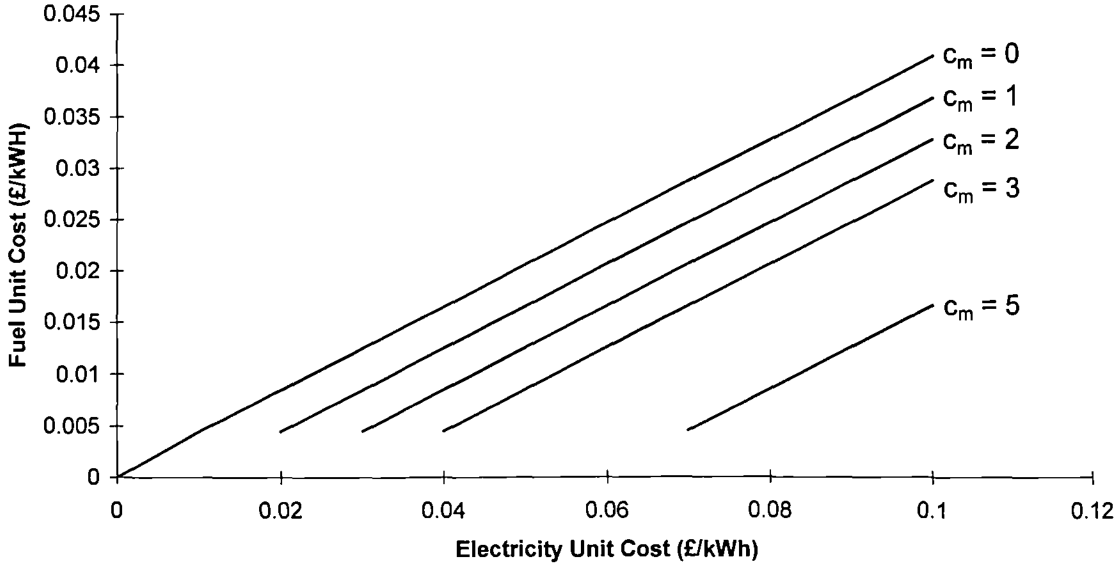


Figure 8.10 Envelope of CHP Economic Operation

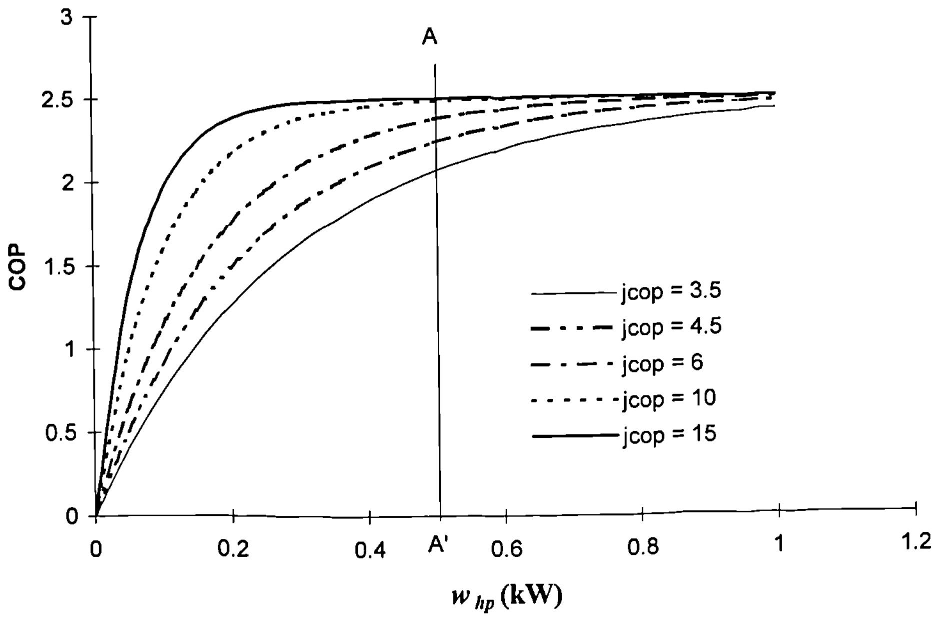


Figure 8.11 Part Load COP With Variable j_{cop}

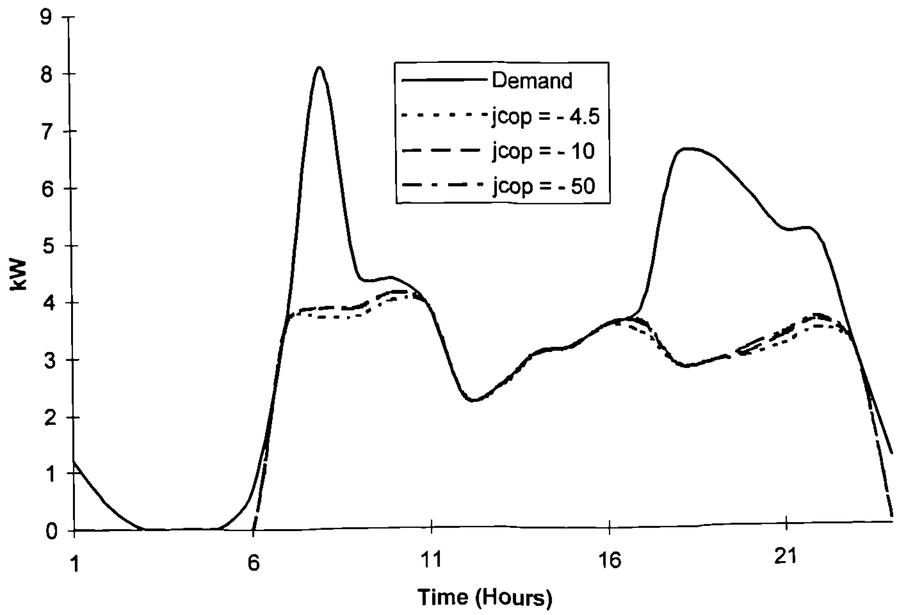


Figure 8.12 Thermal Demand and CHP/HP Deliveries with Variable j_{cop}

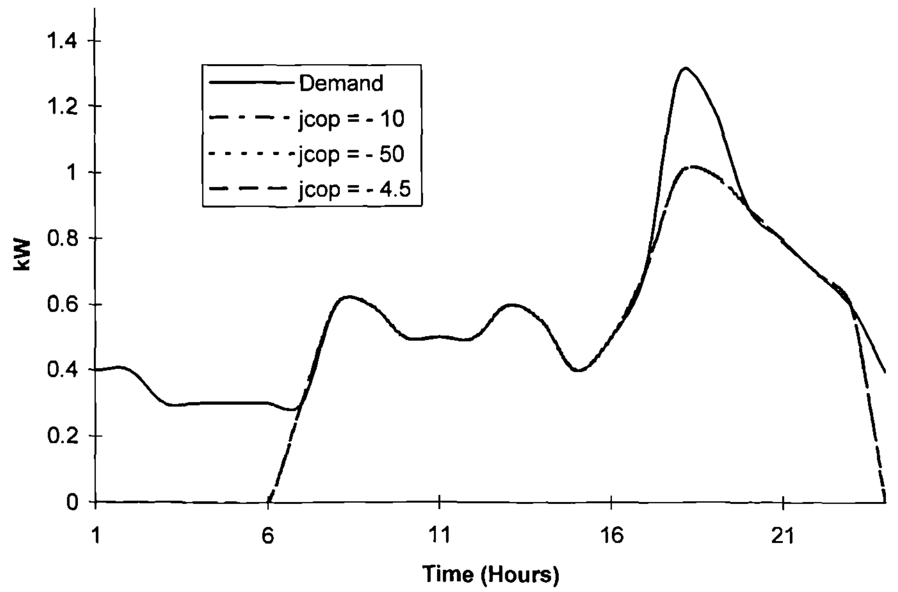
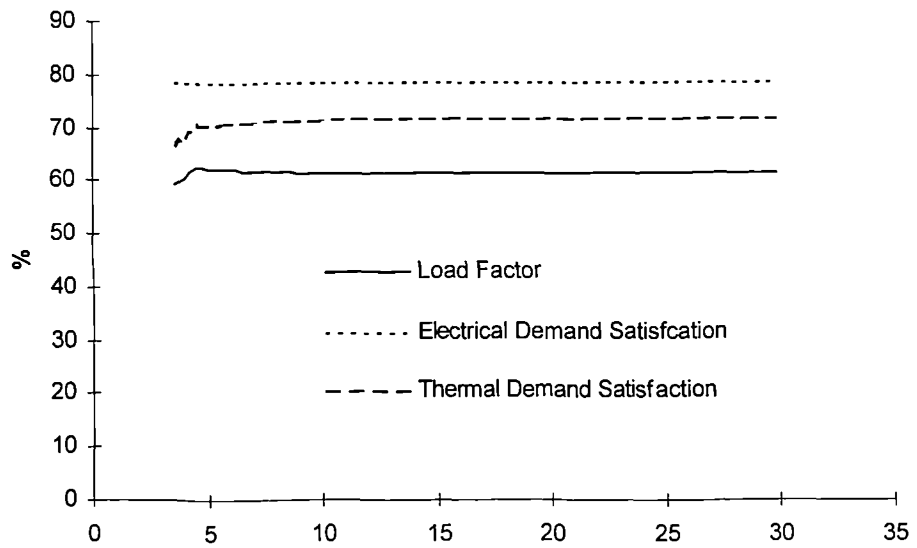


Figure 8.13 Electrical Demand and CHP/HP Deliveries With Variable j_{cop}



j_{cop}

Figure 8.14 Operational Performance of CHP/HP Performance with respect to Variable Heat Pump Part Load Characteristics

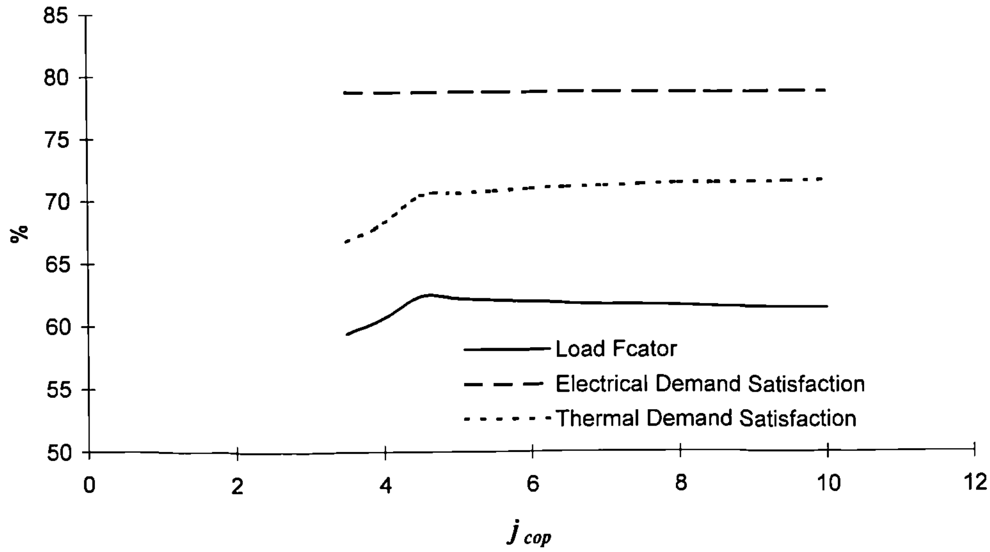


Figure 8.15 Detail of Operational Performance of CHP/HP Performance with respect to Variable Heat Pump Part Load Characteristics

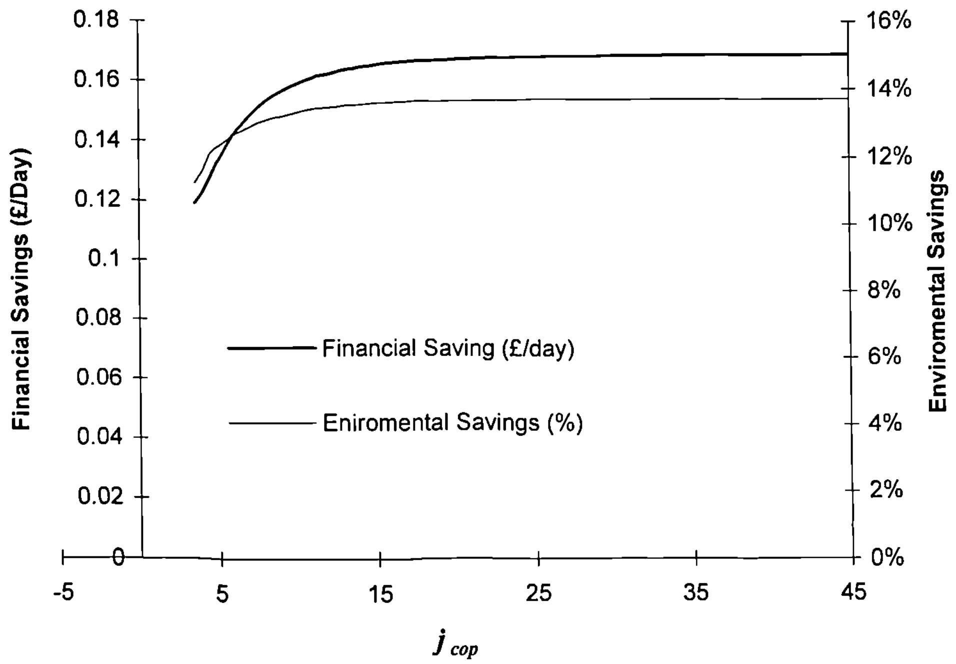


Figure 8.16 CHP/HP Financial Performance with respect to j_{cop}

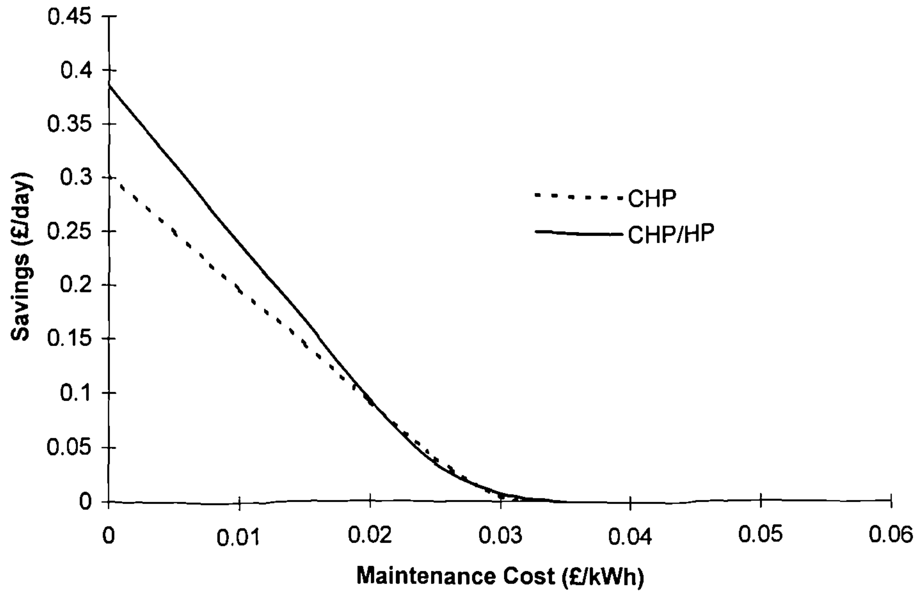


Figure 8.17 Comparison of CHP and CHP/HP Sensitivities to Maintenance Unit Cost

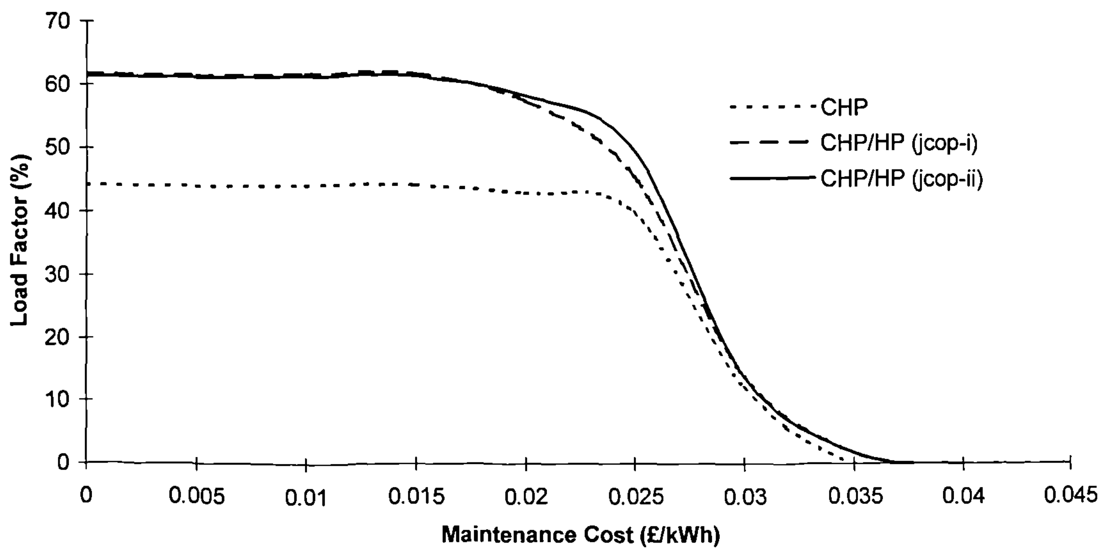


Figure 8.18 Comparison of CHP and CHP/HP Load Factor Sensitivity to Maintenance Cost

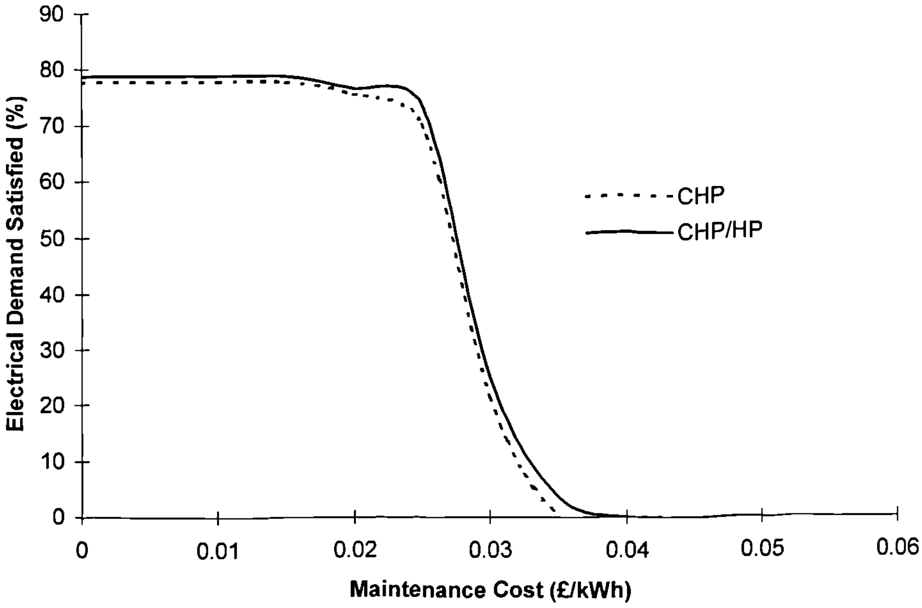


Figure 8.19 Comparison of CHP and CHP/HP Electrical Deliveries with respect to Maintenance Unit Cost

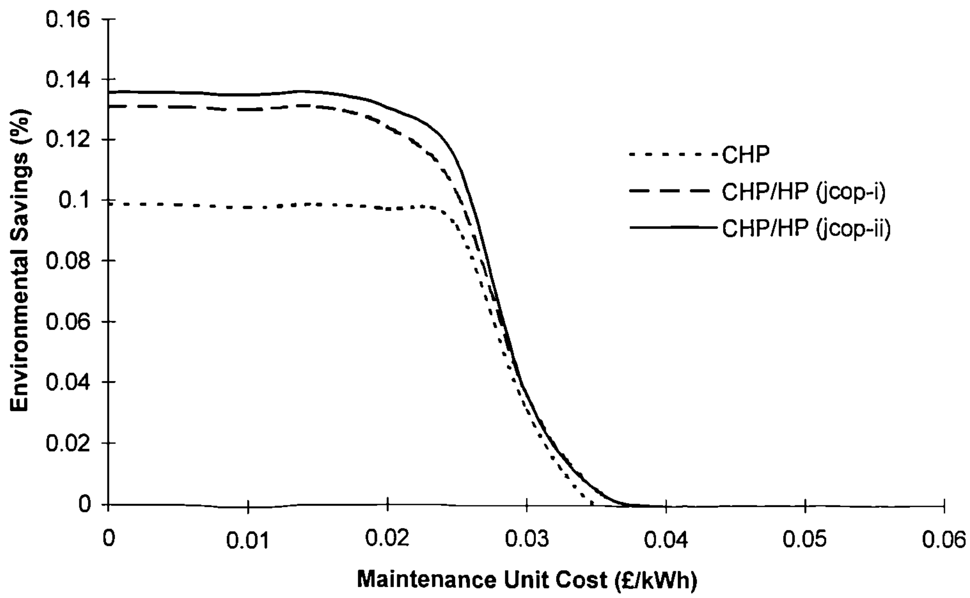


Figure 8.20 Comparison of CHP and CHP/HP Environmental Sensitivity to Maintenance Cost

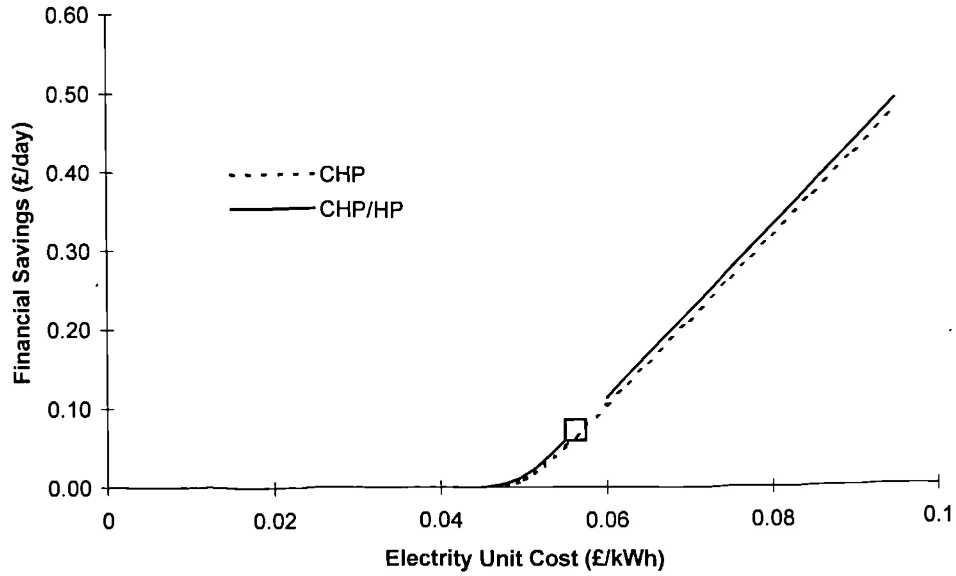


Figure 8.21 Comparison of CHP and CHP/HP Financial Performance with respect to Electricity Unit Cost

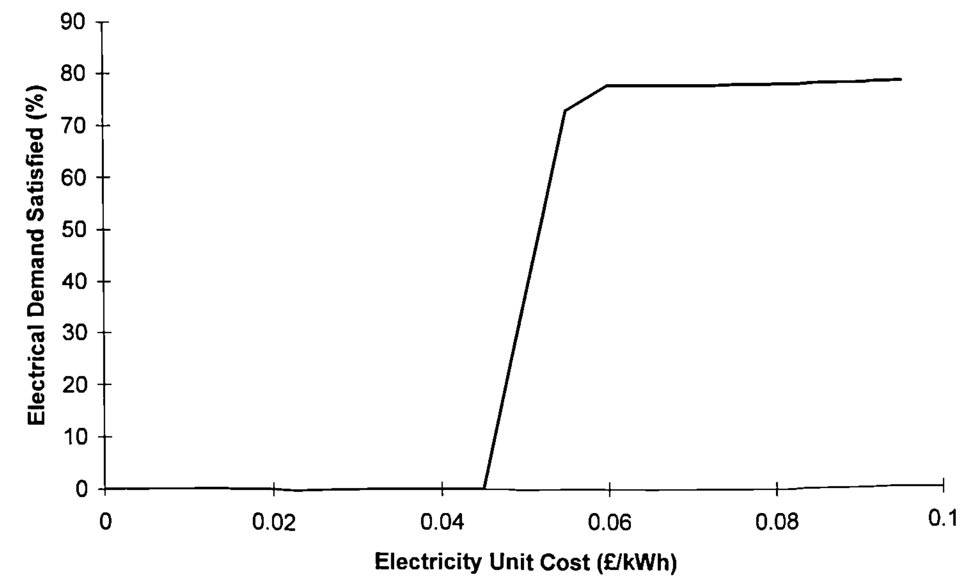


Figure 8.22 Comparison of CHP and CHP/HP Electrical Deliveries with Respect to Electricity Unit Cost

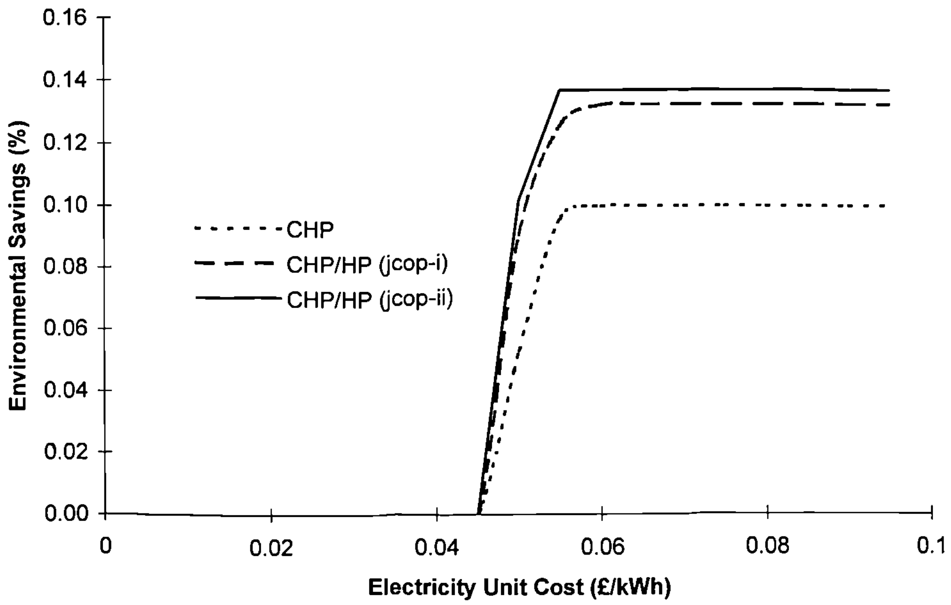


Figure 8.23 Comparison of CHP and CHP/HP Environmental Performance with Respect to Electricity Unit Cost

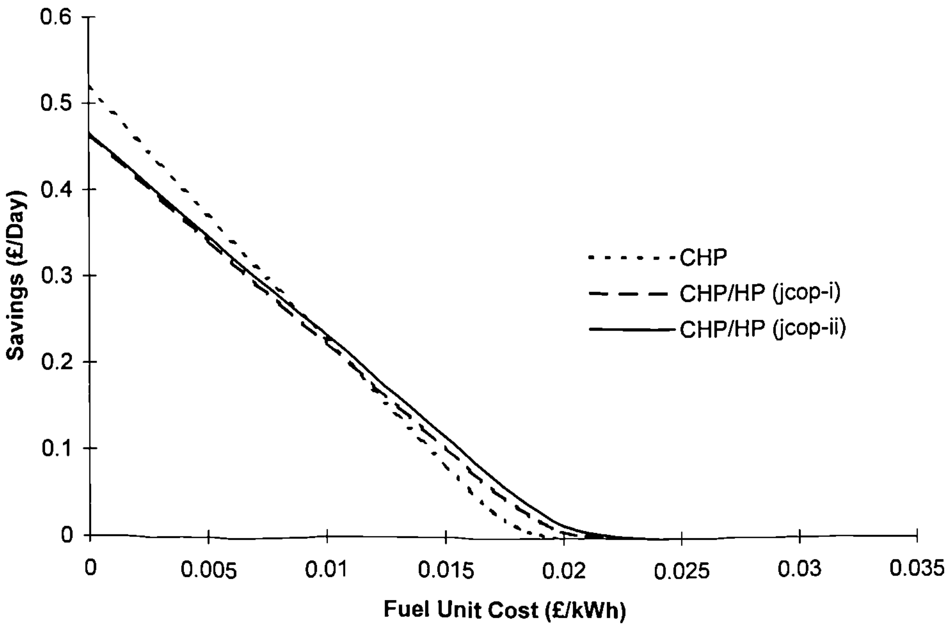


Figure 8.24 Comparison of CHP and CHP/HP Financial Performance with respect to Fuel Cost

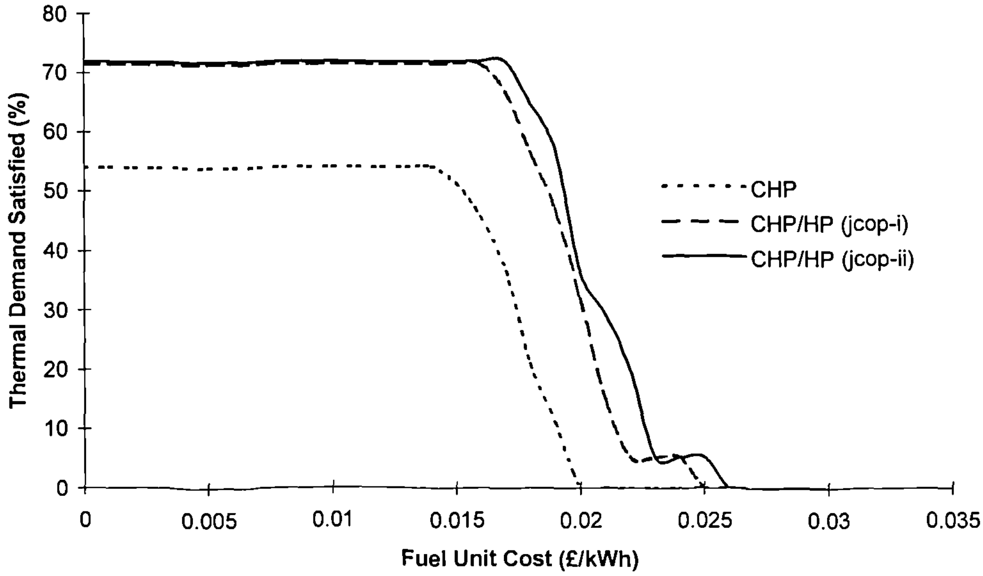


Figure 8.25 Comparison of CHP and CHP/HP Thermal Deliveries With respect to Fuel Unit Cost

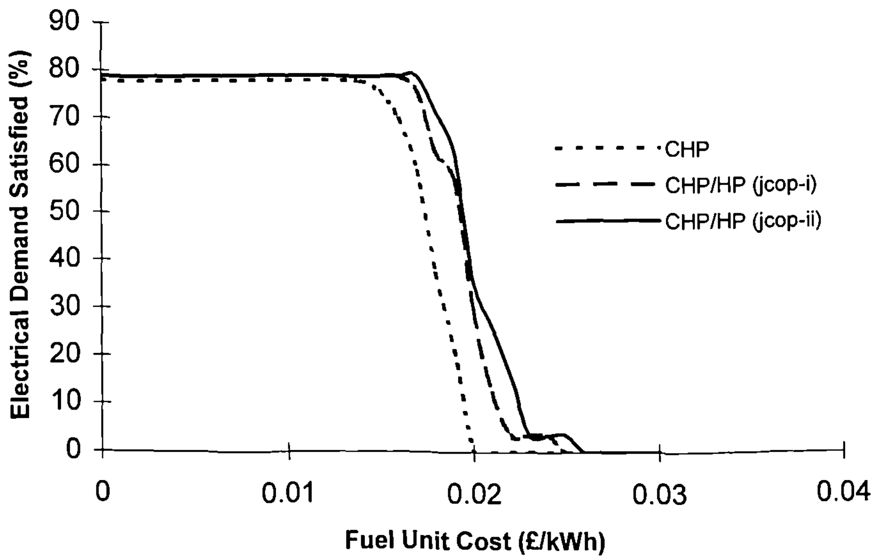


Figure 8.26 Comparison of CHP and CHP/HP Electrical Deliveries With respecf to Fuel Unit Cost

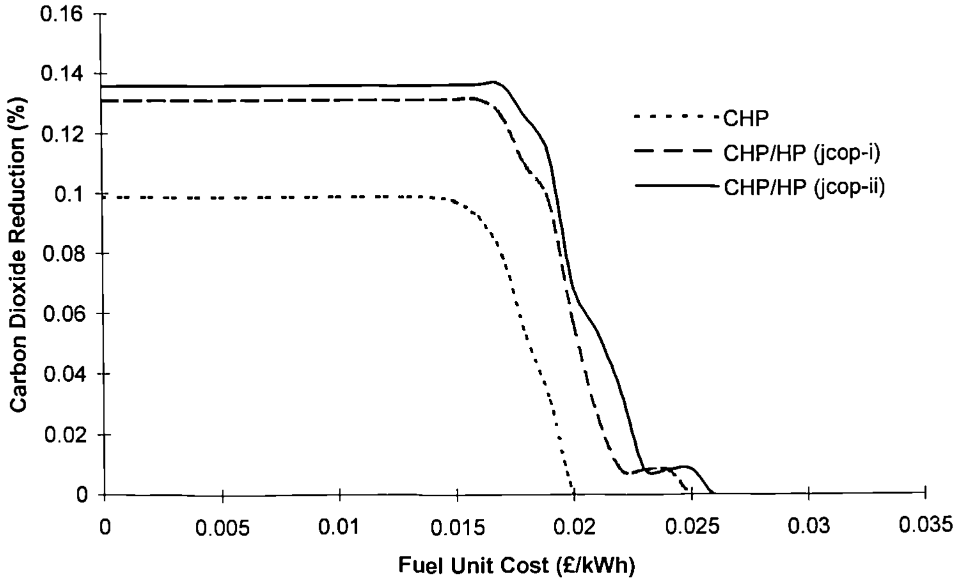


Figure 8.27 Comparison of CHP and CHP/HP Environmental Performance with respect to Fuel Unit Cost

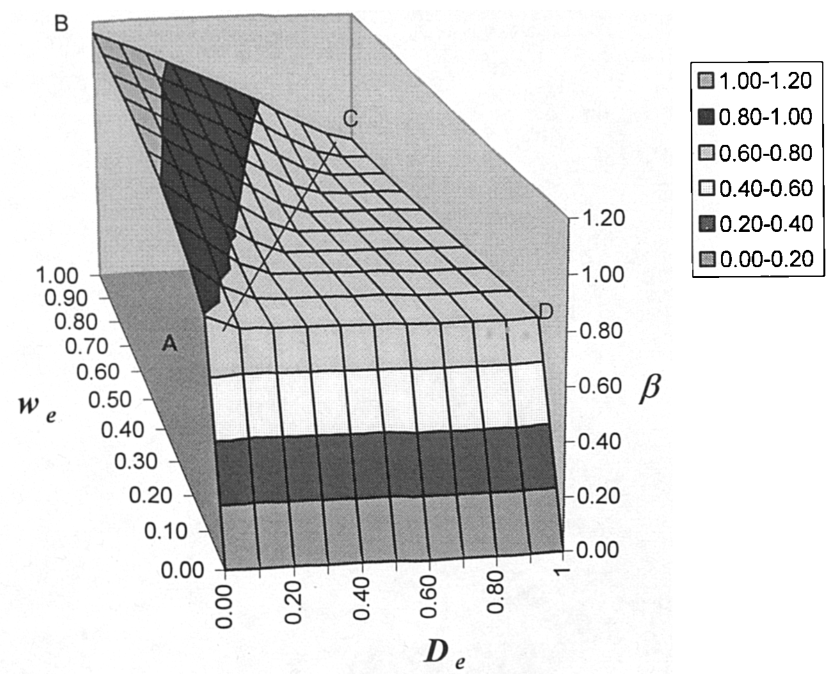


Figure 8.28 Beta Function for CHP/HP Operation

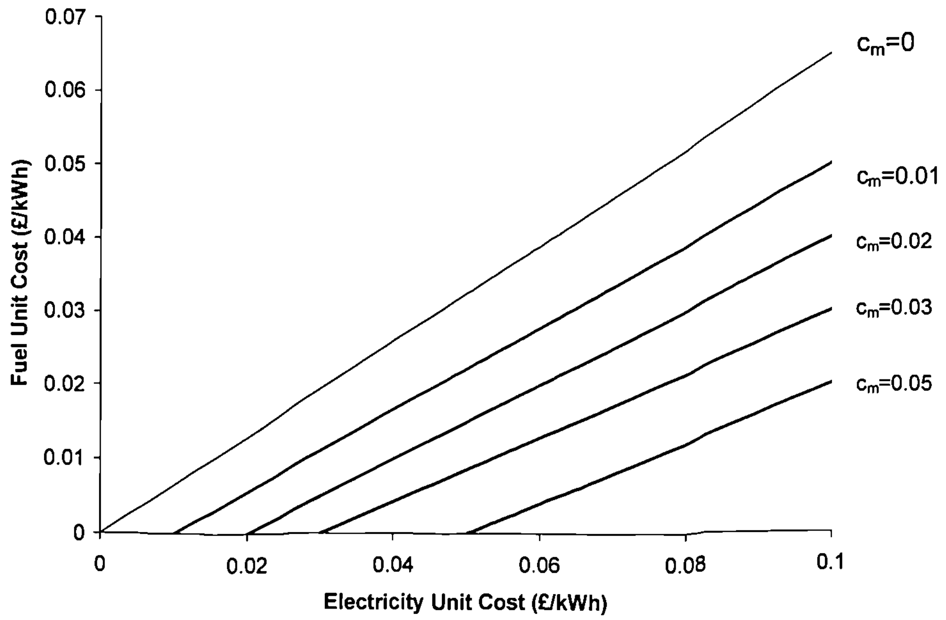


Figure 8.29 Envelope of Economic CHP/HP Operation

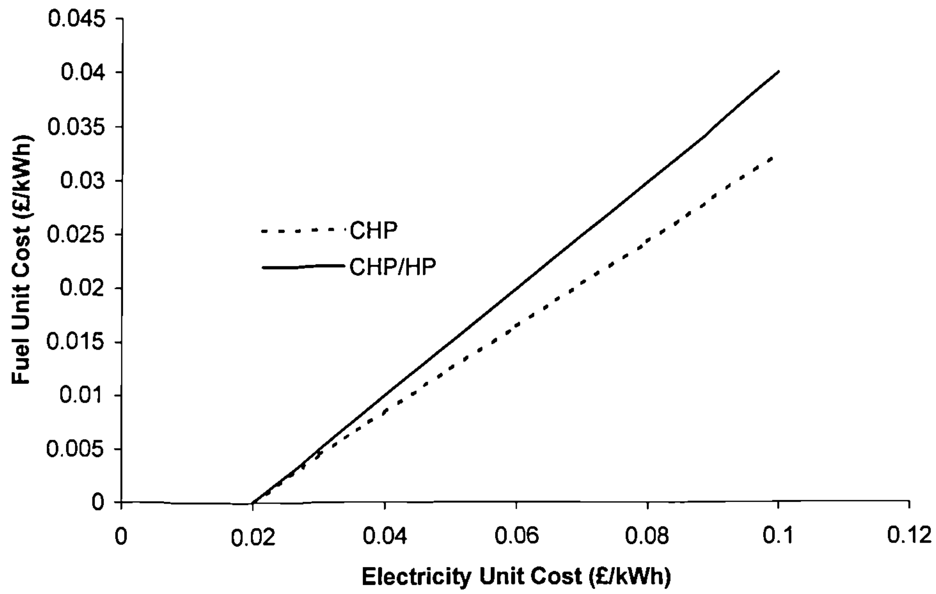


Figure 8.30 Comparison of Envelopes of Economic Operation for CHP and CHP/HP at a Unit Maintenance Cost of £0.02/kWh

9. Conclusions

The following conclusions will draw from both practical and modelled findings and highlight areas of particular importance.

This thesis reviewed existing commercial small scale CHP technology, with a view to domestic application. It was found that conventional CHP plant was inappropriate for domestic application, leading to the development of the CHP/HP concept, which incorporates a heat pump into a CHP plant. Experimental work and subsequent modelling sought to compare the CHP/HP concept with CHP for domestic applications.

Section 9.1 summarises the economic assessment of both types of co-generation. Section 9.2 will compare both domestic scale CHP/HP and CHP in operational and environmental terms. Practical considerations arising from both experimental and modelled work will be discussed in section 9.3. Inferences for future work will made in section 9.4.

9.1 Economic Assessment of Domestic Co-Generation

This section will summarise the economic assessment of both CHP/HP and CHP in domestic application, with respect to plant maintenance and conventional energy costs. A further comparison of economic feasibility between CHP/HP and CHP will be made in section 9.1.5, with respect to varying economic conditions.

Prototype plant costs indicate that the pay-back period would be in the region of 54 years (see Appendix D1.6). However, owing to the experimental nature of the prototype plant, it would be inappropriate to present more specific pay-back periods and NPVs, as the costs incurred in the construction of a prototype plant are not representative of those of a production unit.

9.1.1 Maintenance Unit Costs

Analysis of simulated results shows that maintenance unit costs are critical to economic viability of both types of co-generation examined. It was assumed that heat pump maintenance costs were negligible. Therefore, maintenance costs incurred by both types of operation were due entirely to the engine, and hence identical. With reference to sections 8.2.1 and 8.4.1, it was found that under present economic conditions:

- Maintenance costs must be lower than **£0.032kWe** for domestic co-generation to be viable.

- For domestic co-generation to be effective and make a significant contribution towards domestic energy requirements (and hence have a significant environmental effect) maintenance costs must be lower than **£0.018/kWe**.

A value of £0.015kWe represents a target maintenance value that must be achieved for any practical domestic scale CHP plant to be viable.

9.1.2 Electricity Unit Costs

The majority of financial savings generated with domestic co-generation are as a consequence of displacing utility supplied electricity. With reference to sections 8.2.2 and 8.4.2, analysis found that (with current fuel costs and with a target maintenance cost of 0.015kWh):

- For both CHP/HP and CHP to be economically feasible, the unit cost of utility supplied electricity must be greater than **£0.05kWh**.
- To maximise plant energy delivery and hence environmental advantage, the utility unit cost must be greater than **£0.06kWh**.

9.1.3 Fuel Unit Costs

Table 9.1 summarises the analysis of fuel unit costs (see sections 8.2.3 and 8.4.3). Maximum fuel costs allowing for economic plant operation are presented for both modes of operation. Table 9.1 also presents the fuel unit costs that would optimise environmental and operational performance. These results are subject to the target maintenance unit cost and the current electricity unit cost. CHP/HP can tolerate higher fuel costs than CHP, and in addition, environmental effects are maximised at a higher fuel unit cost.

Table 9.1 Salient Fuel Costs

	Maximum Fuel Cost	**Optimum Fuel Cost**
	£/kWh	£/kWh
CHP	0.016	0.014
CHP/HP	0.021	0.016

9.1.4 Part Load Characteristics

It was found that the part load characteristics of both the engine and heat pump had a significant effect on the cost effectiveness of domestic co-generation. Although engine part load efficiencies were not varied, it is clear that a poor part load efficiency leads to a greater sensitivity to economic variations.

Analysis of the part load COP characteristic showed that a poor part load COP will result in CHP/HP operation being less cost effective than CHP operation. It was found that (with current energy unit costs and the target maintenance cost):

- For CHP/HP to be economically viable, the incorporated heat pump has to have a part load COP characteristic where the associated exponential constant is greater than 15 (see figure 9.1).

- For optimal energy delivery and environmental performance, it is necessary for a heat pump to have a part load COP characteristic with an exponential constant of 8.

9.1.5 Envelopes of Economic Operation

As discussed in section 8.3, heat pump incorporation significantly extends the envelope of economic operation for domestic co-generation (see Figure 8.30). As a variation in electricity unit costs does not favour either CHP/HP or CHP, the extension of the envelope of economic operation is due to heat pump incorporation, allowing CHP/HP to operate at higher fuel costs. However, extremely low fuel costs favour CHP (section 8.4.3).

CHP/HP favours lower maintenance costs compared to CHP. This again highlights the importance of achieving low maintenance costs.

9.2 Comparison of CHP/HP and CHP

This section will contrast the operational and environmental aspects of CHP/HP operation with that of CHP, highlighting and discussing variations.

9.2.1 Operational Performance

With reference to the experimental results and subsequent analysis (see chapter 6), it can be seen in table 9.2 that the first law performance of CHP/HP is significantly better than that for CHP. The electrical efficiency of the CHP/HP case is low, as the *electrical delivery* is a small proportion of engine/generator electrical output: the actual electrical efficiency of the engine/generator is of the order of 15%, but the majority of this output would be utilised by the heat pump. The total efficiency of CHP/HP operation is significantly improved over that of CHP. Total efficiency in this case is effectively a primary energy ratio, which does not take into account the energy acquired by the heat pump from ambient air. As noted in section 6.3.1, total CHP/HP efficiency (not considering heat pump thermal input) may be considered as an economic efficiency, where only the energy inputs that incur an economic cost are considered – i.e. fuel input.

These results physically demonstrate the advantages of heat pump incorporation, allowing the plant to satisfy extremely low electrical demands while supplying the relatively high thermal demands economically. The heat pump work input allows the engine to run at a high load and increases the thermal output of the plant, hence demonstrating that the principle of CHP/HP operation (as stated in section 4.2) is valid.

Table 9.2 First Law Comparison of CHP/HP to CHP Performance

	w_e	Q_{th}	η_e	η_{th}	η_{total}
	kW	kW	%	%	%
CHP	0.98	2.89	15.0	44.3	59.4
CHP/HP	0.22	4.63	3.00	71.0	74.0

Table 9.3 compares the second law performance of CHP/HP operation with that for CHP operation. In contrast to the first law analysis, the second law efficiencies of CHP/HP are poor compared to those for CHP.

Again the electrical delivery for CHP/HP (expressed as the electrical exergy delivery E_e) is low, as the majority of the engine/generator electrical output is delivered to the heat pump – this is reflected in the associated second law efficiency (Ψ_e). The thermal exergy delivery (Ψ_{th}) is greater for CHP/HP operation, as a consequence of the heat pump increasing the temperature of the LPW system flow to the EHE and hence increasing availability – consequently increasing second law thermal efficiency. The total second law efficiency for CHP/HP is significantly lower than for CHP operation, as a consequence of the low electrical delivery associated with the CHP/HP mode. Second law exergy analysis returns a misleading result due to low electrical delivery.

The advantages of heat pump incorporation are highlighted by the exergy analysis of the heat pump operation in both HP and CHP/HP modes (see sections 6.4.2. and 6.4.3). The increase of the exergy delivery from the heat pump to the LPW system is in the order of 250% of what could be achieved by direct heating.

Table 9.3 Second Law Comparison of CHP/HP to CHP Performance

	E_e	E_{th} [1]	Ψ_e	Ψ_{th}	Ψ_{total}
	kW	*kW*	%	%	%
CHP	0.98	0.15	14.5	2.20	16.7
CHP/HP	0.02	0.16	3.20	3.60	6.8

[1] Note: $E_{th} = E_5 - E_2$, represents the exergy delivery from the plant to the LPW system.

9.2.2 Comparison of Environmental Performance

Chapter 8 consistently demonstrates that CHP/HP operation evolves between 30% to 40% less CO_2 than equivalent CHP operation. Heat pump delivered thermal energy displaces the more polluting boiler delivered energy. The financial advantages of heat pump incorporation are marginal in many cases, while the environmental advantages are significant.

9.2.3 Comparison of Models

A comparison of the results of the initial model (described in chapter 4) and those of the validated *concept evaluation model* (developed in chapter 7) show a number of discrepancies. These are due to:

- Assumed part load engine efficiency characteristic used in the initial model.
- A linear relationship for λ used in the initial model – the actual characteristic derived from testing was much improved.
- An ideal heat pump part load COP characteristic used in the initial model.

9.3 Practical Implications

The following section will draw inferences from modelled and experimental results in order to consider the practical implications for domestic co-generation.

9.3.1 Losses Experienced

Both first law and second law analysis of the prototype plant show that the majority of losses are experienced within the engine. First law analysis indicates that 32% of fuel energy input is lost through a combination of radiative and convective heat transfer from the engine cylinder head and cylinder block. Second law analysis reiterates the first law result, with intrinsic losses in the order of 70% of available fuel exergy content. Some of these losses could be recovered by the use of water cooling: however, the transfer of heat to water would still produce large exergy losses, due to the wide temperature differential that would be experienced.

First law analysis of EHE performance indicates that the general design (documented in section 5.5.4) is appropriate, as little improvement could be made. The second law analysis of the EHE returns relatively high exergy losses: these are intrinsic (due to the temperature differential) and hence unavoidable.

9.3.2 Full and Part Load Engine Efficiency

A low full load engine efficiency will have a direct effect on the cost effectiveness of domestic co-generation - where relatively small engines must be employed. Poor part load efficiencies increase the economic sensitivity of domestic co-generation. Hence, for domestic co-generation to be economically feasible for a wide range of conditions, good engine part load efficiency is critical.

9.3.3 Part Load Heat Pump COP

As noted in section 9.1.4, analysis (see section 8.3.2.2) of a poor part load heat pump COP increases the sensitivity of plant cost effectiveness to variation in economic conditions. The importance of heat pump part load COP to environmental performance has been demonstrated and hence a commercial plant must maintain high part load COP. This could be achieved through the use of throttled expansion or an arrangement of set expansion capillaries and associated valves. With both these approaches, control would be an issue.

9.3.4 Electrical System

The problems associated with the use of an AC electrical system were documented in section 6.1.1.1. True domestic implementation would require additional synchronisation with the utility supply. The use of a DC generator and heat pump would overcome the problems highlighted in section 6.1.1.1. A DC system would additionally be compatible with photo voltaic cells. The additional cost of a DC to AC inverter, necessary for utility grid compatibility, would be offset by cost reductions, owing to a simple electrical system and the necessity of power conditioning equipment for an equivalent AC system.

9.3.5 Pre-emptive Control

The development of the concept evaluation model demonstrated the need for a pre-emptive control system. Poor control of a complex CHP/HP plant would result in uneconomic operation. A control system for a domestic application would have to consider the exterior conditions and calculate future thermal requirements while responding to immediate electrical demands. Section 7.4.1 implies that a control system must take account of plant thermal capacitance to achieve economic operation. A control system would have to control engine throttle position and, in the case of CHP/HP, the expansion process of the heat pump. A model of plant and dwelling could be embedded into a domestic CHP/HP control system to ensure effective operation. The concept evaluation model could be used as a basis for this.

9.4 Future Work

Recommendations for future work on domestic scale co-generation are considered in terms of computer modelling and practical research.

9.4.1 Modelling

The validation exercise carried out for the concept evaluation model proved that extending experimental results by computer simulation is reliable. It is proposed that the concept evaluation model is further modified to investigate the potential use of fuel cell technology in domestic co-generation.

Recent developments in low temperature polymer fuel cell research (aimed primarily at the automotive market) may be particularly well suited to domestic co-generation applications. Practical fuel cell electrical conversion efficiencies are in the region of 40%: although the relatively high efficiencies will make domestic co-generation more cost effective, a higher electrical conversion efficiency will reduce the heat to power ratio of the plant. A low heat to power ratio will reduce the thermal output of a fuel cell-based plant and hence the environmental effectiveness will be compromised. Such a low heat to power ratio will result in a low value for the β function (see section 8.2.3 and 8.4.4), with attendant effects on economic sensitivity. In cases where rejected heat from the fuel is used in chemical reformation, little useful heat will be available to meet domestic requirements.

Given the potential low heat output of fuel cell-based CHP plant, heat pump incorporation may be a requirement for domestic co-generation. It is intended that the concept evaluation model be adapted to examine fuel cell-based domestic CHP/HP and assess the need for heat pump incorporation in such systems. Any future modelling will additionally require better data on part load characteristics of heat pumps.

9.4.2 Practical Work

Although computer modelling could assess in broad terms the economic feasibility of domestic fuel cell based CHP/HP, experimental work in a number of areas is necessary for the development of a viable CHP/HP plant.

It has been concluded that a CHP/HP system has significant advantages over a simple CHP plant, and an appropriate heat pump installation with a good part load COP is essential. An incorporated heat pump must be subject to a high degree of control, as must the engine (or fuel cell). It is proposed that a second generation CHP/HP plant be constructed to address these issues. The main focus of future practical research should be in appropriate heat pump design and control. A second-generation prototype should include the following:

- Water-cooled engine (in the absence of a fuel cell), if possible.
- A DC based system, with a DC generator and heat pump compressor motor, allowing for compatibility with fuel cells or PV devices.
- A purpose designed heat pump to examine methods of part load control.
- A pre-emptive control system with an embedded model.

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Appendix

Appendix A. Case Study of a Large Scale Co-generation Plant (Kelenföld, Budapest, Hungary)

A.1 Introduction

This appendix describes the history and operation of a large-scale co-generation plant that provides power and heat to a large urban district. Technical and institutional issues will also be discussed. This appendix will be concluded with a summary that will highlight issues relevant to the thesis.

This work was conducted during an exchange with Budapest Technical University (Budapest, Hungary) during May/June 1996. The exchange was funded and organised by The School of Engineering and Manufacture, DeMontfort University and the E.U. Tempus program.

Sources used were company documents and interviews with power plant engineering staff /management and governmental officials.

A.2 Commercial/Geographic Background

The appendix examines the Kelenföld power plant, situated on the Western (Buda) bank of the Danube, in Budapest (see Figure A.1). This plant supplies heat and power to the south-western districts of Budapest, which are comprised of high density housing, commercial premises and mixed industrial sites.

The Kelenföld power plant is owned and operated by the *Budapest Power Plant Company* (BE Rt.). This is a public/private owed utility formed out of the *Budapest Heat and Power Company*, that supplied energy to the City during the 1900's. BE Rt meets 67% of Budapest's heating requirement with its six power plants (including Kelenföld), supplying 134,000 flats (see Figure A.1 and Table A.1). An annual thermal output of 14.7PJ was required to meet this demand in 1995.

Table A.1 Power Plants of Budapest

		Output		Fraction of total	
Plant	**Date**	**Electrical**	**Thermal**	**Electrical**	**Thermal**
		MW	*MW*	%	%
Angyalfòld	1963	10	62	04	04
Kelenföld	1914	197	503	75	32
Kispest	1962	24	267	09	17
Köbánya	N/A	21	219	08	14
Révész	N/A	0	207	00	13
Ujpest	1912	10	327	04	21
Total		**262**	**1585**		

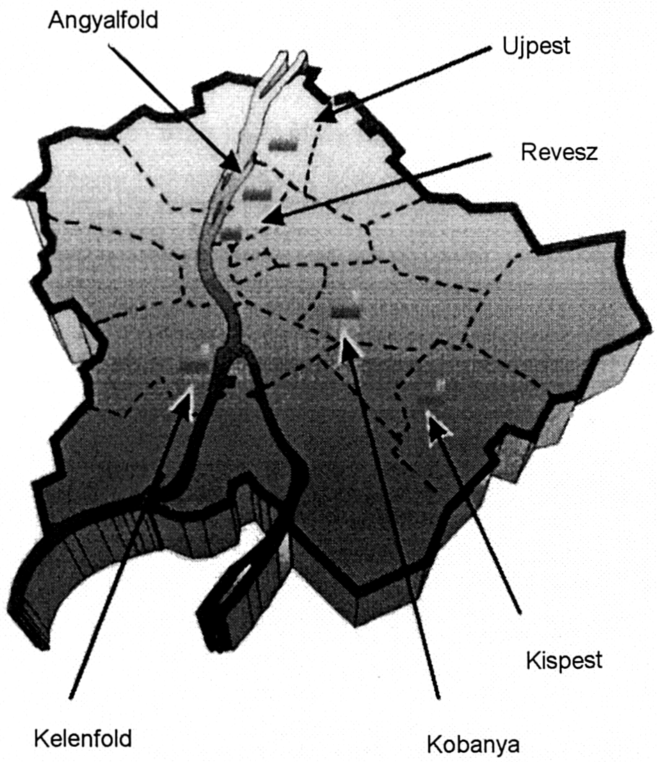


Figure A.1 Budapest Districts and Power Plants

A.3 History of the Kelenföld Co-Generation Plant

The Kelenföld co-generation plant has been developed in three distinct phases:

Early Pre-Soviet Developments – From 1914 to the 1930's, a series of low pressure back-pressure steam turbines were installed, initially fuelled by coal. Three of these units are still in operation (one 5MW unit and two 6MW units, shown in Figure A.1.2.). These units initially had co-generation capacity. The plant at Ujpest is contemporary to this phase of development.

Soviet Developments – During the 1960's (when Hungary was under Soviet control), a number of fully condensing back-pressure steam turbines of a Hungarian design were installed, fuelled by natural gas and oil. Rejected heat from these units is partially used for district heating. Three turbines exist from this period, two 15MW units and one 19MW unit (see Figure A.1.2).

Post-Soviet Investment – In 1995 (after the break up of the Soviet block) BE Rt attracted both Western European investment and contractors. A 137MW gas turbine unit and waste heat boiler (by *ELIN* out of *Westinghouse*) was installed, (see Figure A.1.2), using natural gas and oil as fuel.

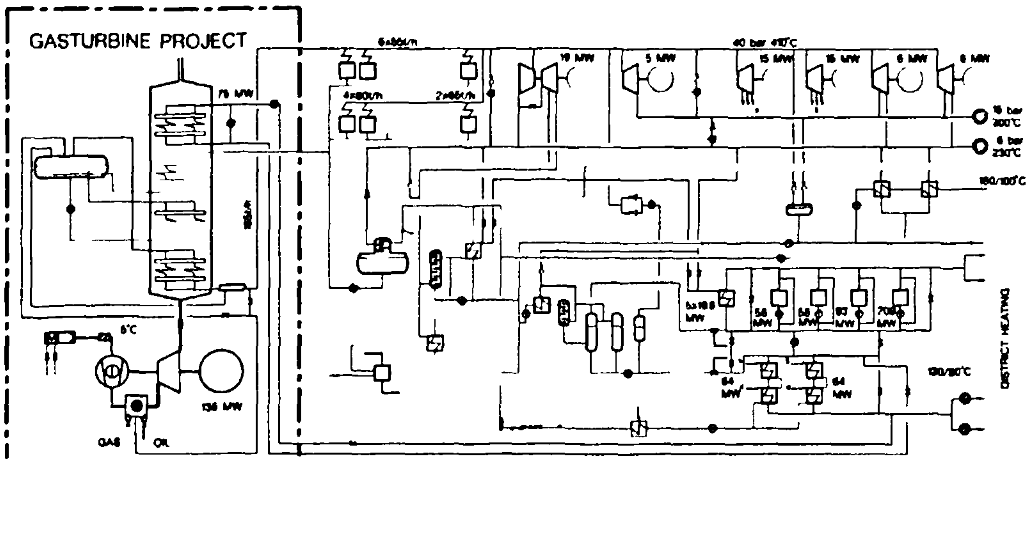


Figure A.2 Schematic of the Kelenföld Co-Generation Plant

A.4 Plant Operation and Management

The gas turbine set is primarily fuelled by natural gas and has an electrical rating of 137MW. The turbine exhaust gas flows through a waste heat boiler, where 196MW of thermal energy is recovered. At this point the gas turbine has a reported fuel conversion efficiency of 84%. 75MW of the recovered heat from the turbine is available directly for district heating, at 80^0C. The remaining 121MW of recovered heat is used to raise steam at 40 bar for use with the steam turbine sets, at a total electrical output of 60MW. Some rejected heat from the steam turbines is used for district heating purposes: 10MW at peak.

The combination of the gas and steam turbine units allow for combined cycle co-generation and single cycle co-generation (when the gas turbine is used independently). The steam turbine units are viewed as unreliable, particularly the Soviet units. Age and spares availability limit steam turbine generation and these units will be decommissioned when alternative generating capacity is installed. Figure A.1.3 summarises the energy flows through the Kelenföld co-generation plant. Figure A.1.3 assumes that all turbo-generators are working at peak and no supplementary steam generation is taking place. These assumptions give rise to a high conversion efficiency for the steam turbine units, although in reality all five steam turbines are seldom used. In practice the gas turbine unit is heavily used and the steam turbine units are used only during periods of peak demand.

Owing to the seasonal variations of residential heat requirements, a thermal output must be either lost to atmosphere in summer or supplemented by conventional boiler plant in winter (Kelenföld has 418MW of boiler capacity). This demonstrates that variation in seasonal residential demand requires substantial redundancy in district heating equipment when large scale co-generation is employed. For example, the facility at Révész has no generating capacity and is purely used for heat supply (see Table A.1).

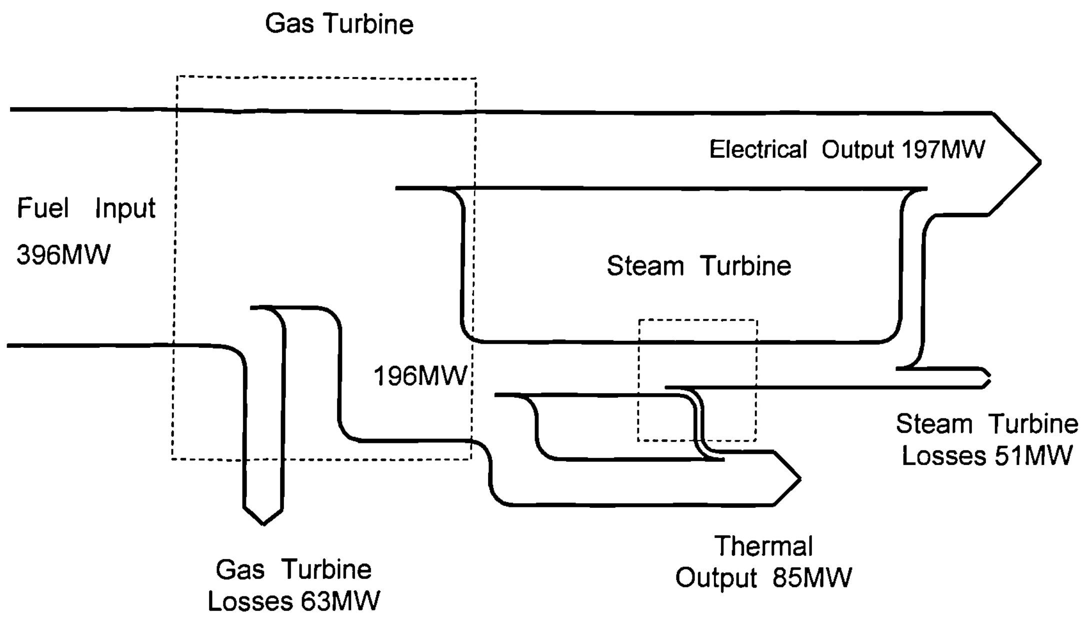


Figure A.3 Sankey Diagram of Kelenföld Co-generation Plant

A.5 Institutional and Political Factors

Although Hungarian energy utilities state that 90% of heating needs are provided by 'co-generation plants', most of the district heating needs are actually met by conventional boiler plant, situated within power plant sites, and not by thermodynamically rejected heat from turbines. The Hungarian Ministry of the Environment officials estimate that 80% of district heating demands are met by conventional boiler plant.

The reliance on conventional boiler plant is a consequence of inexpensive Siberian gas and the importation of inexpensive and inefficient Russian boilers. Lack of investment in co-generation plant has led to the progressive erosion of capacity, due to low cost natural gas and boilers being viewed as an inexpensive alternative.

Recent investments in lucrative co-generation projects have favoured Western technologies, such as at Kelenföld. Small packaged CHP plants (see Section 2.3) have been installed in a number of municipal buildings, such as Budapest Technical University.

A.6 Summary

- Large-scale co-generation schemes supplying the residential sector require a substantial redundancy in boiler plant to meet variations in load.
- The costs of natural gas and associated boiler plant is low enough to marginalise co-generation economics.
- Even when an extensive district heating system exists and large scale co-generation is present, decentralisation by discrete small scale CHP plants is preferable.

Acknowledgements

BE Rt (Budapest Hungary).

Budapest Technical University (Budapest Hungary).

Ministry of Environment and Regional Policy (Budapest Hungary).

Appendix B. Consumption Data

Domestic consumption data used in this thesis were taken from the *Lindford House* project [39] are shown below in Table B.1.

Table B.1 Consumption Data

Time	Demand Thermal (D_{th})	Electrical (D_e)
	kW	kW
00:00	1.20	0.40
02:00	0.40	0.40
03:00	0.00	0.30
04:00	0.00	0.30
05:00	0.00	0.30
06:00	0.70	0.30
07:00	3.70	0.30
08:00	8.10	0.60
09:00	4.50	0.60
10:00	4.40	0.50
11:00	3.90	0.50
12:00	2.30	0.50
13:00	2.50	0.60
14:00	3.10	0.55
15:00	3.20	0.40
16:00	3.60	0.50
17:00	3.90	0.70
18:00	6.50	1.30
19:00	6.50	1.20
20:00	5.90	0.90
21:00	5.20	0.80
22:00	5.05	0.70
23:00	3.10	0.60

C.1 Calculation of Carbon Dioxide Emission

Environmental analysis employed in Sections 3.6, 4.4 and 7.4.7 is derived below. The environmental performance of a co-generation plant considers as a base case the emissions that would be evolved as a consequence of a dwelling's energy demand being supplied from conventional sources. For utility supplied electricity, emissions are evolved for coal, gas (assumed to CCGT) and oil power plants – nuclear plants are ignored. By examining the:

- Specific emissions[1] of each fuel type
- Installed capacity of each type of plant
- Conversion efficiencies of each type of plant
- Utility grid transmission efficiency

The unit mass of evolved emissions per kWh of utility supplied electricity can be calculated, i.e.:

$$p_e = \frac{S_{coal} N_{coal}}{\eta_{coal}\eta_{grid}} + \frac{S_{ng} N_{ccgt}}{\eta_{ccgt}\eta_{grid}} + \frac{S_{oil} N_{oil}}{\eta_{oil}\eta_{grid}} \qquad (C.1)$$

Multiplying the *specific emissions* for utility supplied power with electrical demand gives the emissions due to utility supplied electricity, i.e.:

$$P_e = p_e D_e \qquad (C.2)$$

Similarly, emissions evolved as a consequence of boiler supplied heat can be calculated by examining thermal demand, boiler efficiency and the specific emissions for natural gas, i.e.:

$$P_{th} = \frac{p_{ng} Q_{th}}{\eta_{boiler}} \qquad (C.3)$$

[1] the unit mass of evolved emissions per kWh of fuel consumer at the point of combustion.

Hence, total emissions evolved as a consequence of conventional energy supply are found from combining *C.3* and *C.2*:

$$P_{conv} = p_e D_e + \frac{p_{ng} Q_{th}}{\eta_{boiler}} \qquad (C.4)$$

The emissions evolved as a consequence of a dwelling's energy demand, for a co-generation case, are comprised of those evolved by the co-generation plant and the emissions due to meeting the remaining energy demand (adjusted demand). The adjusted electrical and thermal demands are given by:

$$D_e' = D_e - w_e \quad (C.5) \qquad \text{and} \qquad D_{th}' = D_{th} - Q_{th} \quad (C.6)$$

Calculation of emissions due to co-generation plant use considers the electrical efficiency of the plant, plant electrical output and specific emissions for the fuel:

$$P_e = \frac{w_e S_{ng}}{\eta_{chp}} \qquad (C.7)$$

Substituting the value for adjusted demand into Equations *C.2* and *C.3* and combining with Equation *C.7* gives the total emissions for the co-generation case:

$$P_{chp} = D_e' p_e + \frac{D_{th}' S_{ng}}{\eta_{boiler}} + \frac{w_e S_{ng}}{\eta_{chp}} \qquad (C.8)$$

The reduction in emissions due to co-generation energy supply is found by subtracting *C.8* from *C.4*:

$$P_{conv} = \left(p_e D_e + \frac{p_{ng} Q_{th}}{\eta_{boiler}} \right) - \left(D_e' p_e + \frac{D_{th}' S_{ng}}{\eta_{boiler}} + \frac{w_e S_{ng}}{\eta_{chp}} \right) \qquad (C.9)$$

C.2 Emission Constants

Table C.1 gives the specific emissions for carbon dioxide with respect to different types of energy plant, with associated efficiencies and installed capacities. The source of this data is given below:

Table C.1. Emissions Constants

	Specific Emissions	Conversion Efficiency	Grid Capacity
	Kg/kWh	%	%
Coal	0.3437*	30	40+
Oil	0.3027*	30	10+
CCGT	0.1612	45	20+
Boiler	0.1612	70	

*- Source: Evans[25]

+ - Source: [42]

Grid transmission losses are taken as 10% [25].

C.3 Calculation of Natural Gas (N.G.) Specific Emissions

Table C.2 Data for Combustion Calculation

Description	Symbol	Value	Units	Source
Density of CH_4	ρ	0.554	Kg/m^3	[36]
Calorific Value of CH_4	CV	33.95	MJ/m^3	[40]
Atomic mass of Carbon	C	12	G	[36]
Atomic mass of Oxygen	O	16	G	[36]
Atomic mass of Nitrogen	N	14	G	[36]
Atomic mass of Hydrogen	H	1	G	[36]

Assuming stoichiometric combustion, and that natural gas (N.G.) comprises solely of methane, the combustion mass balance for N.G. is:

$$CH_4 + m[0.21O_2 + 0.79N_2] \rightarrow n_1CO_2 + n_2H_2O + n_3N_2 \qquad (C.10)$$

Solving for:

C: $1 = n_1$ *(C.11)*

H: $4 = 2n_2 \Rightarrow n_2 = 2$ *(C.12)*

O: $0.42m = 2n_1 + n_2$ *(C.13)*

Substituting n_1 and n_2 into *C.13*.

$$0.42m = 4 \Rightarrow m = 9.52$$

Completing mole balance:

$$CH_4 + 9.52[0.21O_2 + 0.79N_2] \rightarrow CO_2 + 2H_2O + 7.52N_2$$

Comparing the mole fractions for CH_4 and CO_2: one mole of CO_2 is evolved for every mole of CH_4 combusted.

Calculating atomic weights:

CO_2: 12 + 2(16) = 44 g/mole
CH_4: 12 + 4(1) =16 g/mole.
Hence, 16g of CH_4 evolves 44g of CO_2.

The volume taken up by 16g of CH_4:

$$V_{CH_4} = \frac{m_{CH_4}}{\rho_{CH_4}} = \frac{0.016[kg]}{0.554[kg/m^3]} = 0.0289[m^3]$$

The energy release from the combustion of 16g of CH_4.

$$Q_{CH_4} = CV_{CH_4}.V_{CH_4} = 33.95[MJ/m^3]0.0289[m^3] = 0.980[MJ]$$

As 1kWh = 3.6MJ:

$$Q_{CH_4}[kWh] = \frac{Q_{CH_4}[MJ]}{3.6[MJ/kWh]} = \frac{0.980[MJ]}{3.6[MJ/kWh]} = 0.272[kWh]$$

The combustion of 16g of CH_4 releases $0.272kWh$ of energy and evolves 44g of CO_2. Hence, the unit mass of CO_2 evolved in the release of 1 kWh of energy is given by (and hence the specific emissions of natural gas):

$$p_{NG} = \frac{1[kWh]0.044[kg]}{0.272[kWh]} = 0.162[kg/kWh]$$

C.4 Calculation of Water Evolution Natural Gas Combustion

Similarly the mass of water evolved can be calculated, by considering n_2 :

$$1CH_4 \rightarrow 2H_2O$$

By considering atomic weights:

16g (of $CH4$) $\rightarrow$ 18g (of H_2O)

Hence, the unit mass of water evolved per kWh of fuel combusted is found from:

$$p_{NG} = \frac{1[kWh]0.018[kg]}{0.272[kWh]} = 0.066[kg / kWh]$$

C.5 Properties of Stoichmetric Exhaust Gas

Table C.3 summarises the calculation of the *cp* and density of the stoichmentric exhaust gas. Data for individual species was taken or interpolated from property tables – [36]

Table C.3 Properties of Stoichometric Exhaust Gas at Different Temperatures

Species	Mols	Mol Fraction	Mol Mass	Mass Fraction	Properties at 300K		Properties at 400K		Properties at 500K		Property at 750K	
					Density	Cp	Density	Cp	Density	Cp	Density	Cp
					kg/m^3	kJ/kgK	kg/m^3	kJ/kgK	kg/m^3	kJ/kgK	kg/m^3	kJ/kgK
N_2	7.52	0.715	210.560	0.725	1.177	1.005	0.882	1.014	0.706	1.030	0.471	1.087
H_20	2.00	0.190	36.000	0.124	0.023	1.864	0.590	1.901	0.460	1.954	0.280	2.114
CO_2	1.00	0.095	44.000	0.151	1.788	0.846	1.341	0.939	1.014	1.073	0.726	1.146
Total	10.52	1.000	290.560	1.000	**1.127**	**1.087**	**0.916**	**1.112**	**0.722**	**1.151**	**0.486**	**1.223**

Appendix D. Additional Prototype Plant Details

This appendix contains additional detail on relating to the development and construction of the prototype plant. Section D.1 details aspects of the prototype plant referenced in this thesis, including technical specifications, component descriptions and development data. Calibration data for individual transducers is held in Section D.2.

D.1 Prototype Plant Specifications and Design Details

The specifications of the engine/generator set are listed in Section D.1.1; LPW system components are detailed in Section D.1.2 and Section D.1.3 describes transducer type and placement. Specifications of the VV204 heat pump are listed in Section D.1.4. A breakdown of prototype plant development costs is shown in Section D.1.5. Section D.1.6 relates gas jet sizes to air fuel ratios, while Section D.1.7 contains development calculations and results for EHE1 and EHE2 designs.

D.1.1 Briggs and Stratton IC3 Specifications

The specification of the engine/generator set is as follows:

Engine - Single cylinder side valve, with 'splash' lubrication.

Generator - Makko single phase 240/110 v 50hz synchronous alternator.

Rated Electrical Output - 1.5kWe.

Fuel - Unleaded petrol.

Ignition system - Fixed induction coil and magnetic fly wheel insert.

Engine Speed Control - Centrifugal governor inside the crank case, operating the throttle via a spring returned linkage.

Stroke = 60mm, **Bore** = 42mm, **Head Volume** = 25ml

D.1.2 LPW System Component Details

With reference to Figure 5.16, the following sections will describe LPW system components.

D.1.2.1 Expansion Unit

A Belco 3.5lt expansion unit was fitted to the LPW system to:

- Allow for the thermal expansion of water within the system.
- Maintain a constant pressure.

A pressure head was required to ensure reliable pump operation. The system was pressurised to run at 1.5 bar.

D.1.2.2 Circulation Pump

A Grundfos UPS2 domestic central heating pump was used to circulate water within the LPW system. The pump was sized to give the optimal flow rate at a system pressure of 1.5 bar.

D.1.2.3 Safety Considerations

As the LPW system constituted a closed pressurised system, a number of safety issues had to be considered. Over-pressurisation of the LPW system would result in the failure of joints, leading to serious equipment damage and possible injury. Over-pressurisation of the system would occur due to insufficient cooling of the water within the system, as a result of equipment failure or a line blockage. The LPW system's safety features are summarised below:

- All pipe work was tested up to a pressure of 3.5 bar.
- Instrumentation was fitted to indicate faults within the LPW system.
- A pressure relief valve was fitted, and set to lift at 3 bar.

D.1.2.4 Pipe Work and Fittings

All pipe work on the LPW system was of 15mm copper tube. Pipe joints were made either with soldered joints or with compression fittings - the latter were used in areas that required frequent modification.

Connections to the EHE were made with steel armoured flexible rubber tubing, to isolate the LPW system from any residual engine vibration. The heat pump sub-system connections were made with rubber hoses secured with pipe clips, so that the heat pump could be positioned independently from the CHP sub-system.

D.1.2.5 Valve Gear

Figure 5.15 indicates the position of individual valves. Specific valve applications and duties are summarised below:

- Quarter turn ball valves (Valves V1 through to V5) were used for EHE and heat pump by-pass loops, for fast operation and visual indication of heat recovery configuration.
- Half turn ball valves were used for LPW system filling, pressurisation and flow rate adjustment (V17 and V18 respectively).
- Air bleeding is via a gate valve (V19) fitted at the highest point of the LPW system.
- The LPW system was drained through a gate valve (V09) fitted at the bottom of the system.

D.1.2.6 Water Treatment and System Filling

To prevent corrosion and fouling of components, the LPW system water was treated with corrosion inhibitor. The glycol based inhibitor also served to protect the heat pump from freezing in extreme conditions. The LPW system was filled by progressive back flushing through different sections of the system.

D.1.3 Transducer Details

D.1.3.1 Temperature Measurement

Temperature transducers used are discussed below in relation to each sub-system.

D.1.3.1.1 LPW System

LPW system temperatures were monitored using CMZ-35 semi-conductor devices for the following reasons:

- Linearity of output over a -30^{O}C to 110^{O}C range.
- Proportional output at 10mV per degree.
- Strong construction.

The placement of the devices was so that heat recovery or rejection from the heat pump or EHE could be calculated. The LM35Z device was accommodated in purpose designed and built thermo-pockets fitted to compression fittings (see Figure D.1).

ITEM	DESCR PT ON
1	CHPHP C1, CARRIER BODY
2	CHPHP_C2, CARR ER CAP
3	CMZ35, TEMPERATURE SENSOR
4	3 CORE SCREED CABLE
5	

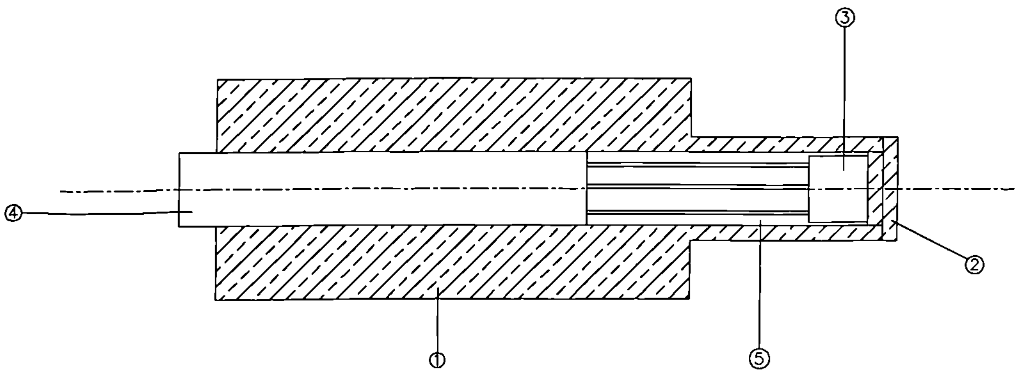


Figure D.1 Thermo-pocket

D.1.3.1.2 Exhaust System

The temperatures were measured either side of the EHE:

EHE Inlet - The high temperature of the exhaust prior to heat recovery required a PT100 platinum resistance thermometer to be used – requiring a double Kelvin bridge to be used to interface the transducer with the ADC system. A thermo-pocket was purpose designed and built for the PT100, as commercial items were too large to be accommodated into the exhaust manifold. The end cap of the thermo-pocket had to be a tight interference fit into the sleeve, as the high temperatures would melt most jointing mediums.

EHE Outlet - The lower temperatures experienced in the EHE exhaust outlet allowed a LM35Z device to be fitted. A thermo-pocket identical to that used for the EHE exhaust inlet was used to accommodate the transducer.

D.1.3.1.3 Heat Pump Temperatures

A LM35Z transducer was attached to the surface of the heat pump compressor, so that an indication of compressor temperatures could be given. Owing to the lack of equipment, no temperature transducers could be fitted to the refrigerant system of the heat pump, as this would have required the evacuation and refilling of the system. However, an appropriate LPW system and air coil temperature would give a good approximation of the refrigerant condition.

D.1.3.2 Pressure Measurement

LPW system pressure - The water pressure within the LPW system was assumed to be constant throughout the system. A single semi-conductor pressure transducer was fitted to measure LPW system pressures. This parameter would have little effect on subsequent analysis and was primarily used as a warning device.

Cylinder Head Pressure - Cylinder head pressure was measured using a piezo-electric device connected to a charge amplifier. Due to high earth leakage associated with the charge amplifier, the transducer was not interfaced with the ADC system as first envisaged, to avoid damaging the ADC circuitry. Therefore, a storage oscilloscope was used to monitor the transducer.

D.1.3.3 Flow Measurement

LPW system - A small turbo-meter (RS 257-149) housed in a compression fitting was used to measure the LPW system flow rate. The meter gave a pulsed output, which was interfaced with the ADC via a frequency/voltage circuit.

Gas Consumption - The gas consumption of the engine was measured with a standard GB1 gas meter, equipped with a pulsed output. By measuring the interval between pulses, the gas consumption could be calculated.

D.1.3.4 Electrical Measurements

Electrical measurements were made using Hall effect devices originally constructed for electric vehicle research by [41]. The voltages and currents were measured for both the heat pump and engine. By calculating RMS values, power could be derived.

D.1.4 Heat Pump Specification

The technical data relating to the Versatemp VV204 heat pump (manufactured by Temperature Limited, Southampton, UK) is shown below.

RATING CONDITION - SI UNITS

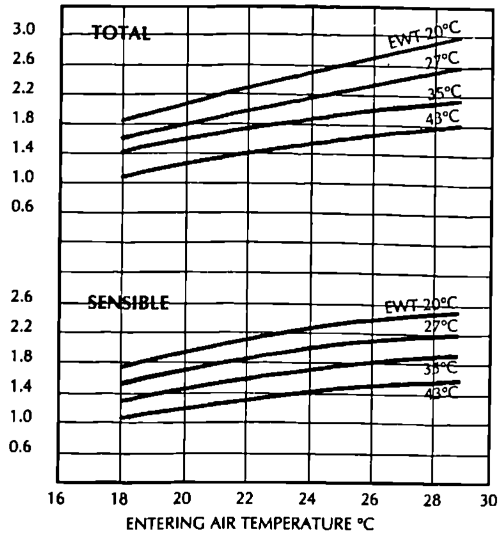
COOLING	Total Output (EAT 24°C 50% RH EWT 35°C) Sensible Output Total Electrical Input	1.85 kW 1.70 kW 0.90 kW
HEATING	Output (EAT 19°C EWT 35°C) Total Electrical Input	3.07 kW 0.78 kW

All performance data is valid for 240/1/50 Hz, med. fan speed and is subject to standard tolerances.

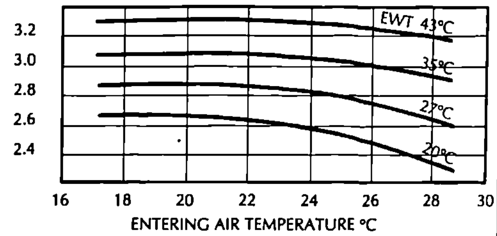
WATER DATA

Flow Rate	0.064 l/s
Pressure Drop	13.8 kN/m²
Content (including hoses)	0.35 lt
Maximum Working Pressure	1380 kN/m²
Max. No of units per 32 Amp Supply	5
Recommended Fuse Rating to BS 1362	13 Amps

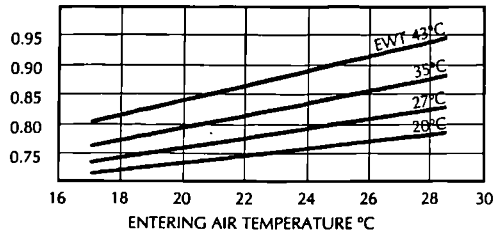
COOLING OUTPUT - kW



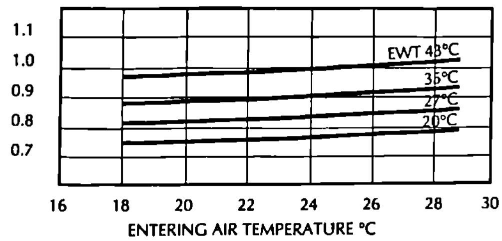
HEATING OUTPUT - kW



HEATING INPUT - kW



COOLING INPUT - kW



OTHER DATA

Full Load Current *(maximum continuous operation, EAT 29°C, EWT 45°C)*	4.58 Amps
Starting Current - 240V	19.8 Amps
Fan Motor Input *(maximum)*	0.10 kW
Free Area for Return Air Grille	0.066 m²

Correction factors for alternative fan speeds	COOLING						HEATING					
	High		Medium		Low		High		Medium		Low	
Voltage at 50 Hz	240	220	240	220	240	220	240	220	240	220	240	220
Total Output	1.02	1.01	1.00	0.99	0.92	0.91	1.04	1.00	1.00	0.96	0.98	0.94
Sensible Output	1.05	1.02	1.00	0.97	0.92	0.89	n/a	n/a	n/a	n/a	n/a	n/a
Total Electrical Input	1.01	0.94	1.00	0.93	0.98	0.91	0.97	0.92	1.00	0.95	1.05	1.00
(Airflow in m³/s)	0.18	0.16	0.15	0.13	0.12	0.10	0.18	0.16	0.15	0.13	0.12	0.10

For 25% glycol/water solution by volume pressure drop increase factor is 1.18 Other factors are:-		Total Output	Sensible Output	Electrical Input
	Cooling	0.92	0.93	1.03
	Heating	1.01	n/a	1.02

Maximum permissible external resistance permissible against low fan speed to achieve published airflow. 0 N/m²

D.1.5 Prototype Plant Development Costs

Table D.1 shows the component and labour costs for the construction of the prototype plant. Labour costs are only quoted for work carried out commercially and not by the author.

Table D.1 Costs of Prototype Plant

		Costs (£)		
Item	**Component**	**Labour(£)**	**Materials(£)**	**Total(£)**
1	**Engine / Generator**	000.00	300.00	300.00
2	**Heat exchanger**	020.00	050.00	070.00
3	**Heat Pump**	000.00	800.00	800.00
4	**Exhaust system**	010.00	040.00	050.00
5	**Chassis & sub frames**	020.00	010.00	030.00
6	**Pump**	000.00	030.00	030.00
7	**Valves**	000.00	050.00	050.00
8	**Pipe work**	020.00	020.00	040.00
9	**Control system**	020.00	150.00	170.00
10	**Transducers**	000.00	050.00	050.00
11	**Vibration mounts**	000.00	030.00	030.00
	Misc. Assembly	020.00	000.00	020.00
		£110.00	£1530.00	**£1640.00**

Labour cost = £20/man hour

With reference to Figure 8.16 daily savings for CHP/HP operation (for the assumed conditions) are £0.165/day. Assuming a 6 month heating season the pay back period for a CHP/HP plant (identical to the prototype plant) would be 54 years.

D.1.6 Gas Jet Sizes

Table D.2 relates gas jet sizes for the converted carburettor to air/fuel ratio. Figure D.2 identifies relevant dimensions of the natural gas fuelling annulus (see Figure 5.4 for the location of this component within the fuel system).

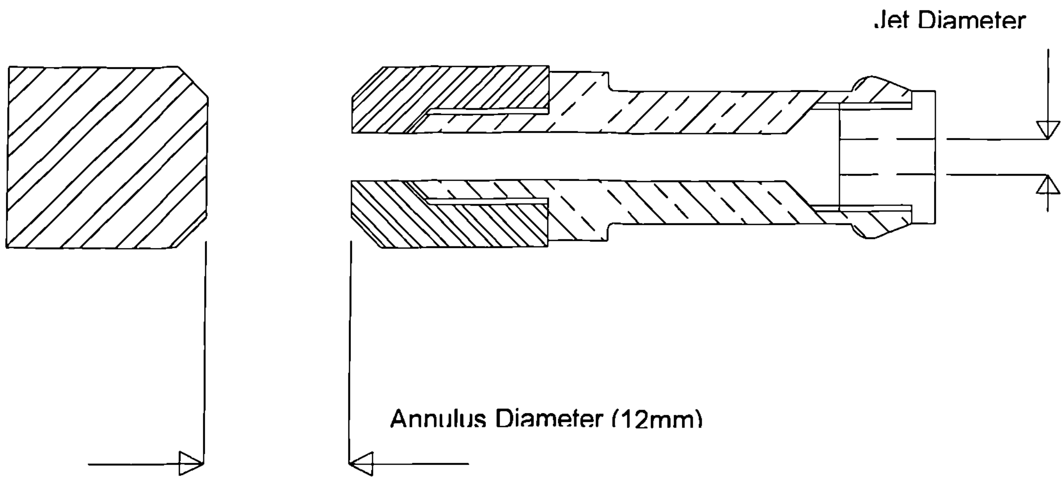


Figure D.2 Gas Fuelling Annulus and Jet

Table D.2 Jet Sizes and Air/Fuel Ratios

Annulus		
Diameter	*mm*	12.00
Area	*mm²*	113.09
Jet Diameter	**Jet Area**	**Air/Fuel Ratio**
mm	*mm²*	
3.20	8.04	14.06
3.50	9.62	11.76
3.75	11.04	10.24
4.00	12.57	09.00

Air/Fuel Ratio = (Venturi Area) / (Jet Area)

D.1.7 EHE1 Performance

The following section presents the performance test results for the EHE1. Figure D.3 shows the temperatures and flow rates during the EHE1 assessment test.

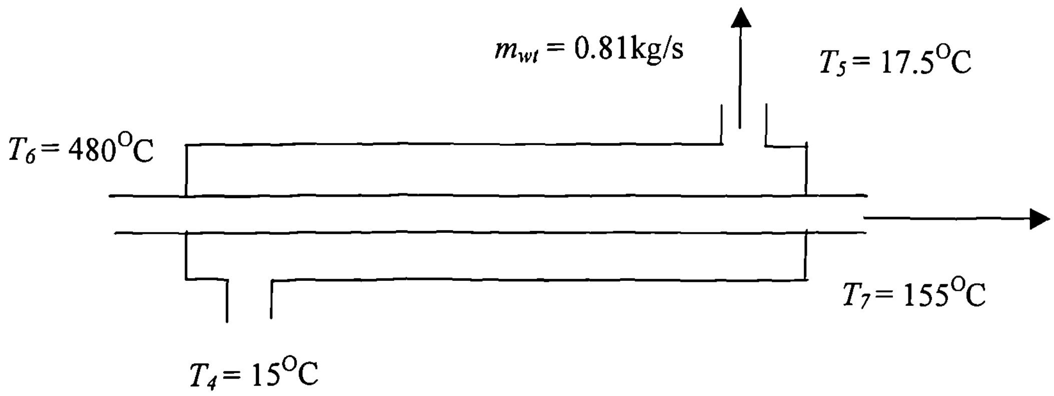


Figure D.3 EHE1 Test Results

Heat delivery from the EHE1 (Q_{ehe}) is calculated from:

$$Q_{ehe} = m_{wt} cp_{wt} (T_5 - T_4) \tag{D.1}$$

Applying data gives: $\boldsymbol{Q_{ehe} = 0.85}$**kW**

Heat input into the EHE1 from the exhaust gas (Q_{ex}) is calculated from:

$$Q_{ex} = m_{ex} cp_{wt} (T_6 - T_7) \tag{D.2}$$

Where the mass flow of exhaust gas ($\dot{m}_{ex}$) is calculated by considering the mass flow of fuel and air into the engine ($\dot{m}_{engine}$) – as the mass entering the engine is equal to the mass leaving the engine, i.e. $\dot{m}_{engine} = \dot{m}_{ex}$

Volume of engine cylinder and head (V_{engine}) is calculated from:

$$V_{engine} = \pi\left(\frac{b}{2}\right)^2 s + V_{head} \qquad (D.3)$$

Where: Cylinder bore (b) = 0.06m
Piston stroke (s) = 0.047m
Cylinder head volume = $2.5\text{x}10^{-5}$ m^3

$V_{engine} = 1.579\text{x}10^{-4}$ m^3

Volume flow per unit time ($\dot{V}_{engine}$) is found by applying the engine speed (50Hz – see Section D.1.1) and dividing by two - as the engine is a four stroke machine, the total volume is recharged every two rotations, hence:

$$\dot{V}_{engine} = \frac{50.V_{engine}}{2} \qquad (D.4)$$

$\dot{V}_{engine} = 3.947\text{x}10^{-3}$ m^3

As the mass of fuel entering the engine is negligible, compared to the mass of air, and the properties of oxygen are similar to those of nitrogen, it will be assumed that the density of gas entering the engine (at room temperature 300K) is that of nitrogen [36]. Hence:

$$\dot{m}_{engine} = \rho\dot{V}_{engine} \qquad (D.5)$$

$\dot{m}_{engine} = \dot{m}_{ex} = (3.947\text{x}10^{-3}).(1.177) = 4.65\text{x}10^{-3}$ kg/s

Applying data to Equation D.2 and using the Cp calculated for 500K in Section C.4):

Q_{ex}= $(4.65\text{x}10^{-3})(1.151)(325)$ = **1.74kW**

The effectiveness (ε) of the EHE1 is calculated from:

$$\varepsilon_{ehe} = \frac{Q_{ehe}}{Q_{ex}} \qquad (D.6)$$

ε_{ehe}= **0.49**

D.1.8 EHE2 Performance

Table D.3 reports the performance of the EHE2 design during its assessment test. The engine was run at full load until stable thermal conditions were achieved, for a number of varying water flow rates. The methods of calculation and assumptions used were those employed in Section D.1.7.

Table D.3 EHE2 Performance

m_{wt}	T_4	T_5	T_6	T_7	*LMTD*	Q_{ex}	Q_{ehe}	ε
kg/s	^oC	^oC	^oC	^oC	^oC	*kW*	*KW*	
0.009	22.00	53.20	480	37.10	70.80	2.52	1.15	0.46
0.017	21.50	44.90	480	34.20	62.43	2.54	1.62	0.64
0.020	21.00	40.50	480	34.70	53.17	2.54	1.60	0.63
0.020	21.00	41.50	480	34.50	55.70	2.54	1.72	0.68
0.026	21.00	36.10	480	33.20	45.60	2.55	1.67	0.66
0.028	20.50	36.60	480	33.20	47.17	2.55	1.91	0.75
0.036	20.50	33.70	480	32.70	37.56	2.55	1.98	0.78
0.044	20.50	30.30	480	32.20	41.69	2.55	1.80	0.70
0.048	20.50	30.80	480	31.70	36.77	2.56	2.08	0.81
0.055	20.00	29.30	480	31.70	43.53	2.56	2.15	0.84
0.065	20.50	28.30	480	31.70	46.48	2.56	2.13	0.83
0.075	20.00	26.40	480	30.30	47.81	2.56	2.02	0.79
0.138	20.00	23.90	480	27.30	46.52	2.58	2.26	0.88

D.1.9 EHE2/3 Core Design

Figure D.4 shows the EHE2/3 core, the plan view is shows the EHE core position within the water jacket.

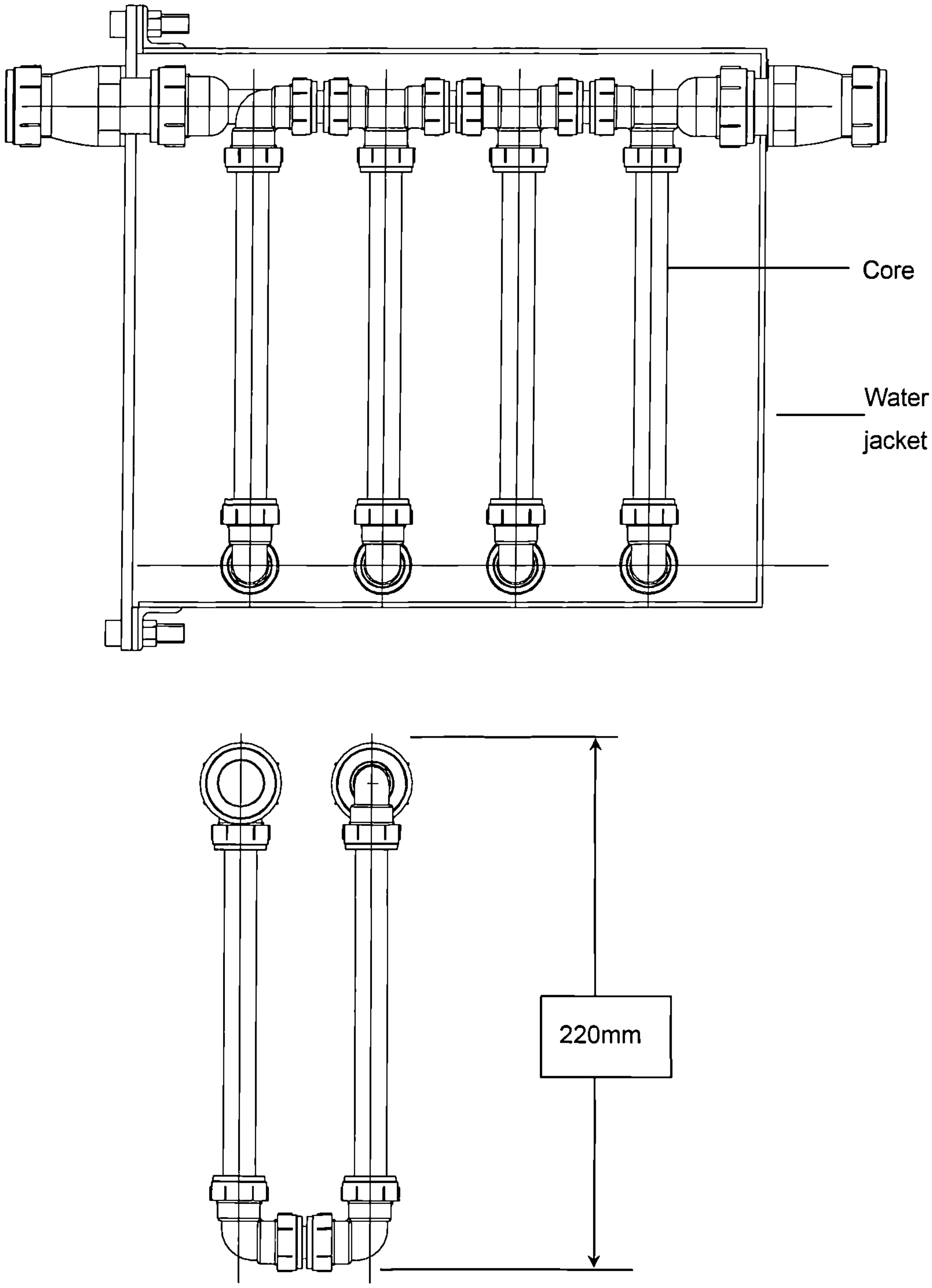


Figure D.4 EHE2/3 Core Design

D.2 Transducer Calibration

This sub-appendix will document the calibration and subsequent analysis of the transducer used in the experimental tests pertaining to this thesis. The calibration and analytical procedure is similar for all transducers:

- The signal output (voltage) of the transducer is compared to the measured reference quantity.
- A conversion function is derived so that transducer input signals can be converted, by the data acquisition system, into the desired units.
- The converted transducer results are compared to measured results, providing statistical data to assess the accuracy of the transducer and for subsequently used sensitivity analysis. All transducer tolerances are subject to a 95% confidence interval.

All transducers were calibrated via the data acquisition system, which takes into account transducer lead/connector resistance and ADC interface characteristics. The following sub-appendices will document calibration and analysis of the transducers in the following order:

- Temperature transducers.
- Flow transducers.
- Electrical measurement transducers.

Summary tables for all calibrated transducers can be found at the end of this appendix.

D.2.1 Temperature Transducer Calibration and Analysis

The calibration of the low temperature LM35Z transducer (excluding the engine exhaust transducers - T_6) was carried out in two stages:

- A single transducer was calibrated using a precision pt100 instrument.
- The remaining LM35Z transducers were then calibrated using the sample transducer.

The numbered index of a temperature transducer refers to the thermodynamic point used in analysis. Temperature transducer calibrations and validations are summarised in Table D.4. The statistical indicators return satisfactory values for all examined temperature transducers.

The values for Δ_m given in Table D.4 are the respective uncertainties for each transducer. The uncertainty on the intercept component of the validation function is negligible and hence ignored. The transducer uncertainty (or the estimated standard deviation of the gradient constant) was calculated using the MS Excel (V97) – Regression analysis tool. These values will be employed in Appendix E.2, along with associated sensitivity analysis, to calculated tolerances on results presented in Chapter 6.

D.2.1.1 Calibration of Sample Temperature Transducer

The sample LM35Z temperature transducer was placed in an oil bath with a precision PT100 digital thermometer. The oil bath was heated and then the temperature was allowed to fall, whilst the oil was constantly stirred. The output from the LM35Z sample transducer was recorded (in mV) by the data acquisition system at incremental reference temperatures obtained using the precision PT100 device.

A linear regression was carried out on the two sets of results to derive a conversion function for the transducers. The conversion function was applied to the recorded transducer signal and the result was compared to the reference reading. Carrying out a linear regression on transducer-derived temperatures, with respect to reference temperatures, allows for statistical validation analysis to be carried out for the transducer.

D.2.1.2 Temperature Transducer Calibrations

The temperature transducers (LM35Z T2 to T10) were placed in an oil bath with the calibrated LM35Z T-1. The oil bath was heated and allowed to cool, while throughout this process the data acquisition system recorded the signals from all the transducers.

The signal from the reference LM35Z T-1 transducer was used to calculate the temperature of the oil bath serving as a reference for calibration of the remaining transducer.

Table D.4 Temperature Transducer Calibrations and Validations

	Calibration			Validation				
Transducer	m	C	r^2	m	c	r^2	Δm	Test
	v/°C	*°C*		*V/°C*	*°C*			
T-1 (ref)	098.909	-0.626	0.999	0.999	0.653	0.999	0.0239	Accept
T-2	090.852	4.001	0.997	0.997	0.092	0.997	0.0168	Accept
T-3	087.706	4.348	0.996	0.996	0.144	0.996	0.0119	Accept
T-4	087.061	4.350	0.996	0.996	0.120	0.996	0.0135	Accept
T-5	104.240	-1.144	0.997	0.997	0.105	0.997	0.0204	Accept
T-7	095.463	2.420	0.999	0.999	0.049	0.999	0.0150	Accept
T-8	095.264	1.520	0.997	0.997	0.090	0.997	0.0226	Accept
T-9	093.375	1.189	0.999	0.999	0.050	0.999	0.0121	Accept
T-10	103.440	-1.454	0.999	0.999	0.033	0.999	0.0115	Accept
T-11	088.766	4.048	0.997	0.997	0.101	0.997	0.0194	Accept

D.2.2 Voltage and Current Transducer Calibration

The calibration procedure for voltage and current transducers was dependant on the application, i.e. engine/generator set or heat pump. Voltage and current transducer calibrations and validations are summarised in Table D.5. Transducer uncertainties are calculated as in Appendix D.4.

With reference to Table D.5, all examined transducers returned satisfactory statistical indicators. The regression coefficient for the I_{hjp} transducer is of a relatively low order, due to the narrow range of reference voltages recorded. The transducer is still reliable, as the voltages measured during testing were of the same order as those used in the calibration.

D.2.2.1 Engine/Generator Set Electrical Transducers

To obtain the calibration of the engine/generator current transducer, the engine/generator was run subject to a resistive electrical load placed on the generator. The I_{gen} transducer signal was recorded along with an independent measurement of generator current (using ISO-Tech IDM205: 22800544). The engine load was varied *so that a range of values could be recorded. Figures D.54* and D.55 present both calibration and validation results.

Additionally the RMS voltage output of the generator was also recorded with the transducer signal from V_{gen} transducer, so that calibration and validation could be carried out for this transducer.

The variation in engine speed for different loads allowed for a range of voltages to be examined (it is assumed that variations in engine speed, and hence AC output frequency and voltage had no effect on the current transducer).

D.2.2.2 Heat Pump Electrical Transducers

The heat pump cycled whilst the RMS supply voltage and current were recorded with respect to the transducer signals (V_{hp} and I_{hp}). As the heat pump was supplied by mains electricity, the variation in voltage is smaller than for the engine/generator set - the heat pump was mains powered during the experimental testing.

D.2.3 LPW System Flow Transducer

The LPW system flow transducer produced a pulsed output, where frequency was proportional to flow rate. A frequency to voltage converter circuit was employed to convert the pulsed signal into a voltage that could be read by the ADC card within the data acquisition system. Both flow transducer and frequency to voltage converter circuits are subject to errors. The calibration of the flow transducer also implicitly considered the frequency to voltage conversion circuit. Initially the frequency to voltage converter circuit was manually adjusted so that the voltage output signal was approximately proportional to the flow rate through the transducer (i.e. 0.6V output signal at 0.06lt/s).

The passage of a volume of water through the flow transducer and the voltage signal from the frequency voltage converter was recorded. This procedure was repeated for a range of flow rates that were experienced in the LPW system during plant operation.

The water flow was provided by the mains supply and reference measurements were made using a 5lt measuring cylinder and stop watch. This test was carried out during the late evening so that localised pressure changes in mains water supply were eliminated.

The statistical evaluation of the LPW system flow transducer is presented in Table D.5. Statistical analysis returns satisfactory indicators.

Table D.5 Summary of the Calibration and Validation of Power and LPW System Flow Transducers

	Calibration			Validation				
Transducer	m	c	r^2	m	c	r^2	Δm	Test
Vgen	74.852	-5.765	0.9992	0.9992	00.012	0.9992	0.0217	Accept
Igen	02.628	-0.185	0.9997	0.9997	00.001	0.9997	0.0122	Accept
Vhp	82.859	-30.527	0.7406	0.7406	64.567	0.7406	0.2170	Accept
Ihp	02.533	-0.020	0.9999	0.9999	00.000	0.9999	0.0042	Accept
LPW Flow	00.0886	0.0024	0.9708	0.9703	00.0017	0.9708	0.0108	Accept

Appendix D.3 Additional Figures of Completed Prototype Plant

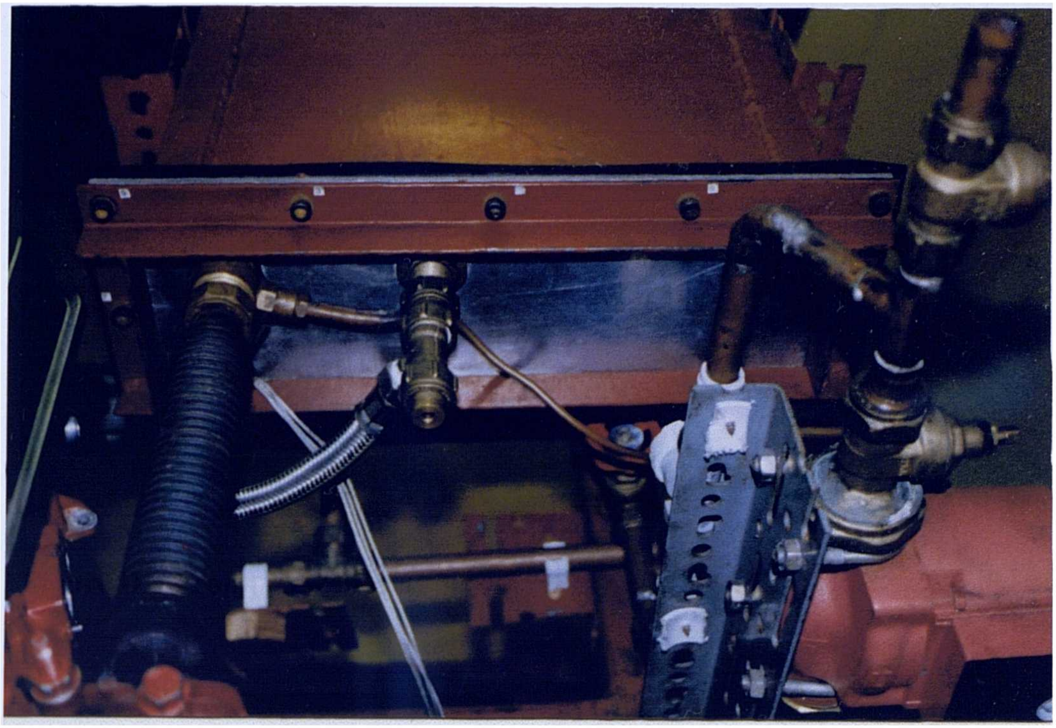


Figure D.5 Front View of EHE3

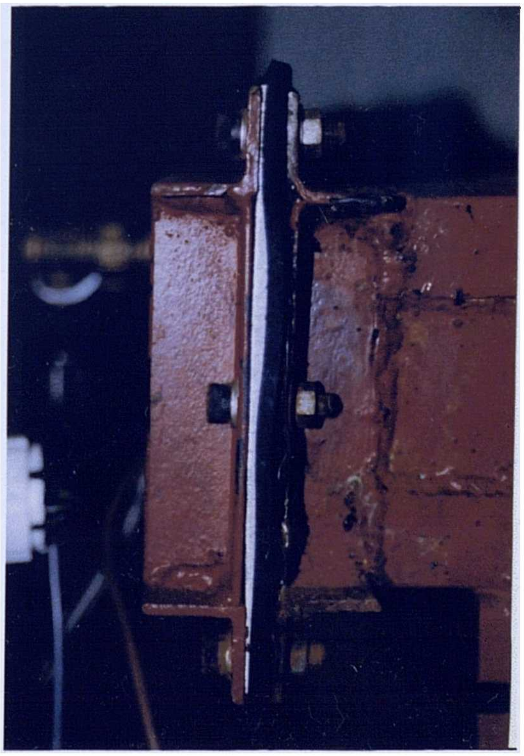


Figure D.6 Side View of EHE3

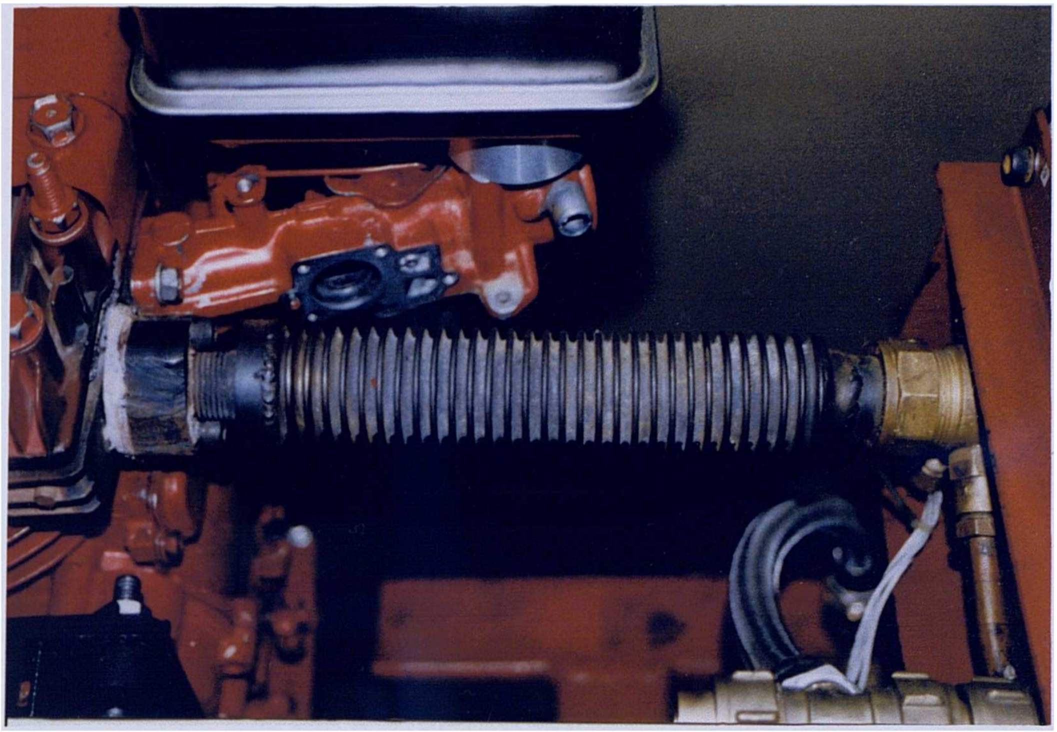


Figure D.7 Flexible Exhaust Link Fitted to EHE3 and Engine

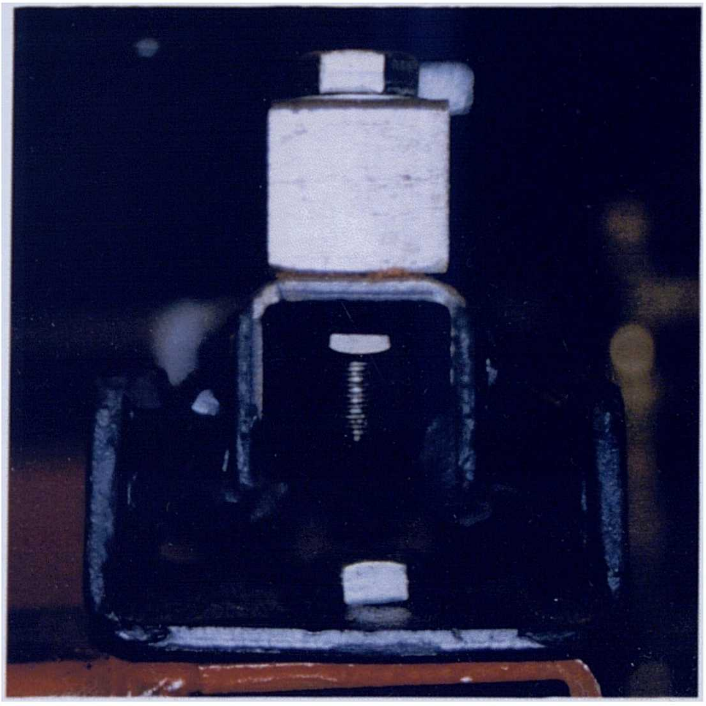


Figure D.8 Double 'U' Mount

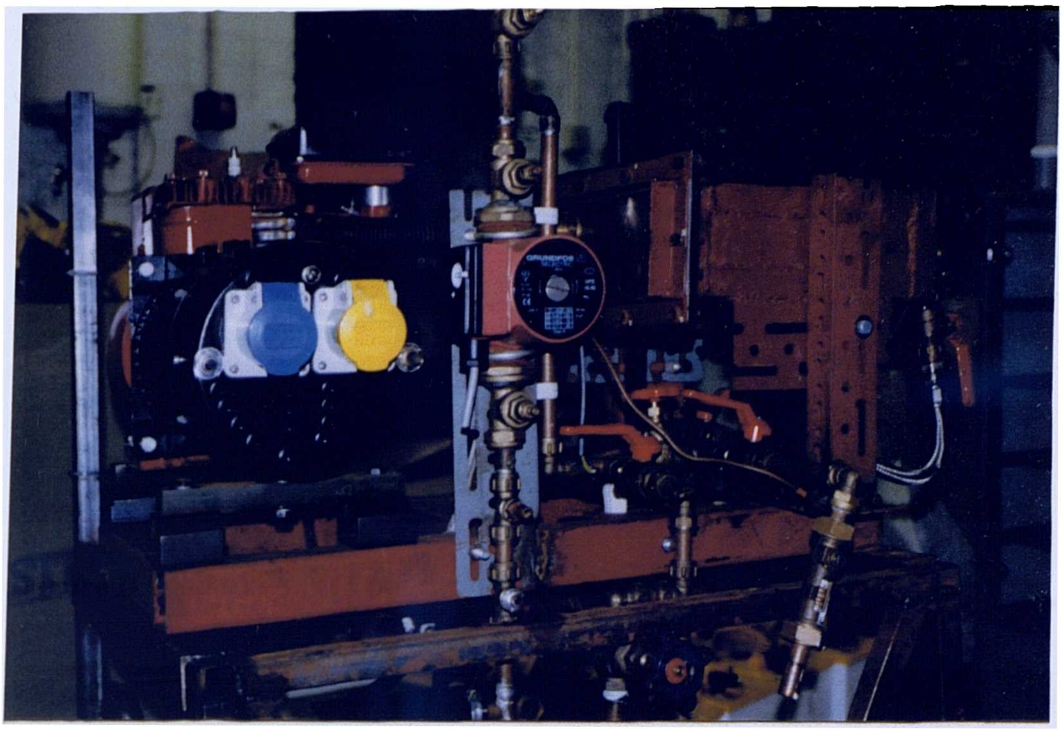


Figure D.9 Side View of Completed Assembly

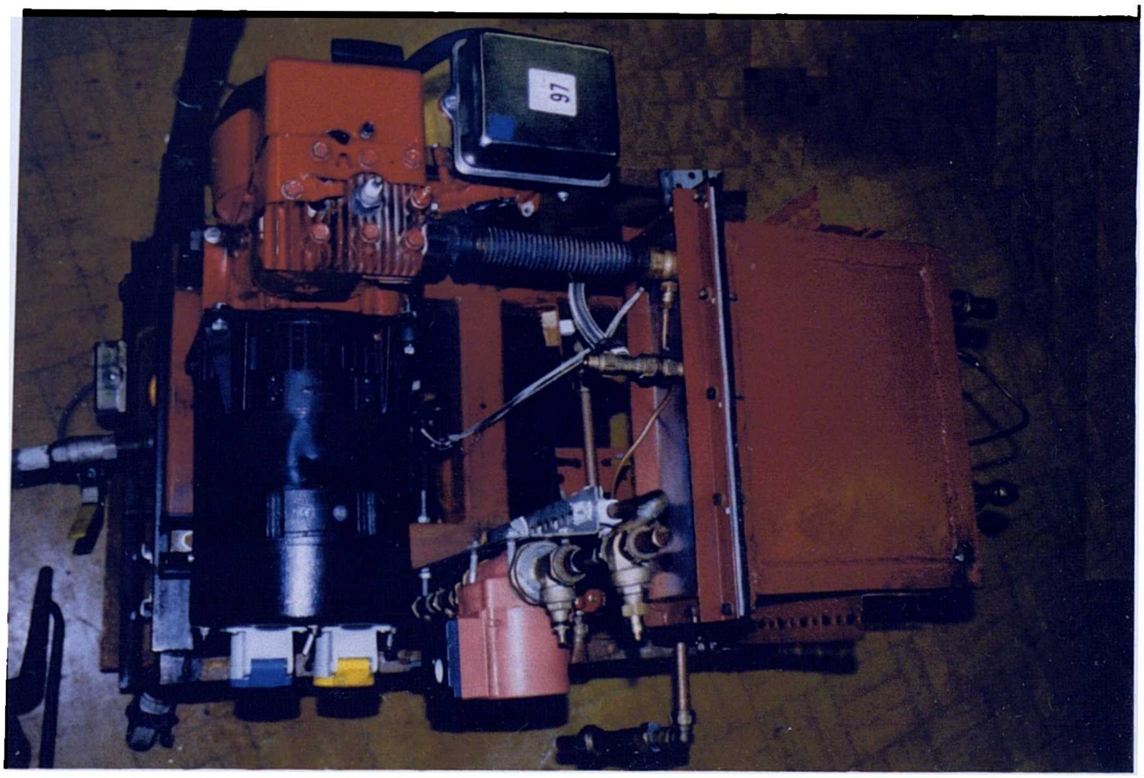


Figure D.10 Top View of Completed Assembly

Appendix E. Test Results and Related Calculations

This appendix will present experimental test data relevant to this thesis (see E.1). Appendix E.2 develops relevant sensitivity analysis and is combined with transducer tolerances (see D.2) in sub-appendix E.3 to support results presented in Chapter 7.

E.1 Test Result

This sub-appendix contains the tabulated results for tests used in this thesis. These tests are summarised below.

CHP Test 1 (see Appendix E.1.1) – The engine was run at full load to overcome thermal capacitance and then reduced. CHP Test 1 was in:

- **Section 6.2.1.** Derivation of engine/generator electrical efficiency characteristics.
- **Section 6.2.2.** Derivation of EHE thermal delivery characteristics.
- **Section 6.3.1.** First law analysis of CHP operation (at time index 11:00:17).
- **Section 6.4.1.** Second law analysis of CHP operation (at time index 11:00:17).
- **Section 7.5.3.1.** Validation of fuel consumption simulation.
- **Section 7.5.3.2.1.** Validation of EHE rapid thermal response.

The negative values for Q_{hp} and hence COP are due to transducer errors (well within the limits defined by sensitivity analysis: see Appendix E.3.1.).

Heat Pump Test - no engine (See Appendix E.3.2) – The heat pump was run without the engine/generator, and results used in the examination heat pump COP in Section 6.2.3. As the engine/generator set was not in operation, the thermal capacitance of the EHE affected the COP. As cold water entered the heat pump heat exchanger, the apparent thermal delivery increased until the thermal capacitance of the heat pump heat exchanger was overcome. This mechanism gave rise to very high values for Q_{hp} and *COP* when water from the EHE was introduced into the heat pump (time index 13:05:43 to 13:24:38).

HP Mode Test (see Appendix E.1.3) – The prototype plant was configured so that the electrical output of the engine/generator set matched the electrical demand of the heat pump, hence the plant effectively ran in HP mode (see Section 4.3.2.). The HP mode test was used in:

- **Section 6.3.2.** First law analysis of HP operation (at time index 15:36:53).
- **Section 6.4.2.** Second law analysis of HP operation (at time index 15:36:53).

CHPHP Test 1 (see Appendix E.1.4) – The engine/generator set was run at full load and the heat pump operated normally, allowing for an effective electrical generator surplus. The anomaly occurring at time indexes 13:38:48, 13:41:28 and 13:44:09 was due to a temperature transducer fault, which was corrected. The CHPHP Test 1 was used in:

- **Section 6.3.2.** First law analysis of CHP/HP operation (at time index 14:02:55).
- **Section 6.4.2.** Second law analysis of CHP/HP operation (at time index 14:02:55).

CHP Test 2 (see Appendix E.1.5) – The engine/generator set was run at full load for a sustained period, then shut off and allowed to cool. CHP Test 2 was used in model validation (see Section 7.5.3.2.2).

CHPHP Test 2 (see Appendix E.1.6) – The engine/generator set was run at full load throughout the test whilst the heat pump was initially run, then turned off and finally restarted at the end of the test. This test was used in model validation (see Section 7.5.3) as opposed to the CHPHP Test 1, to avoid the temporary transducer fault.

E.1.1. - CHP Test 1

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
10:41:35	06.56	28.19	27.81	26.66	29.09	350.48	32.96	31.41	31.69	0.082	0.28	239.27	0.28	0.07	3.53	0.02	0.24	0.32	0.84	238.84	1.26	0.30	-0.13	-0.43	0.93
10:44:25	07.74	29.32	28.93	30.34	34.71	430.89	43.16	32.70	33.03	0.083	0.31	235.79	1.83	0.43	6.33	0.07	0.24	0.29	1.53	233.60	1.35	0.32	-0.14	-0.44	1.75
10:46:58	08.25	29.87	29.47	33.53	39.64	482.64	53.28	33.44	33.70	0.084	0.34	234.08	4.10	0.96	7.04	0.14	0.31	0.35	2.15	234.10	1.34	0.31	-0.14	-0.45	2.13
10:49:32	08.13	29.78	29.36	35.60	42.40	487.22	55.39	33.43	33.65	0.084	0.37	234.18	4.51	1.06	6.97	0.15	0.34	0.39	2.40	234.16	1.34	0.31	-0.15	-0.47	2.43
10:52:07	08.17	29.79	29.37	36.92	44.28	490.33	56.84	33.54	33.76	0.084	0.39	234.04	4.96	1.16	6.94	0.17	0.37	0.42	2.60	233.87	1.34	0.31	-0.15	-0.46	2.56
10:54:47	08.49	30.20	29.74	37.84	45.94	492.72	58.23	33.74	33.96	0.084	0.41	233.68	3.90	0.91	6.71	0.14	0.43	0.47	2.86	233.80	1.34	0.31	-0.16	-0.51	2.73
10:57:32	08.22	29.94	29.47	38.10	46.17	485.26	57.75	33.49	33.69	0.084	0.41	234.18	3.75	0.88	6.53	0.13	0.44	0.48	2.84	234.34	1.34	0.31	-0.16	-0.52	2.66
11:00:17	**08.06**	**29.83**	**29.36**	**38.35**	**46.55**	**485.87**	**58.04**	**33.34**	**33.51**	**0.084**	**0.42**	**234.41**	**4.19**	**0.98**	**6.53**	**0.15**	**0.44**	**0.49**	**2.89**	**234.49**	**1.34**	**0.31**	**-0.17**	**-0.53**	**2.77**
11:03:11	08.01	29.78	29.31	38.39	46.63	484.47	58.19	33.21	33.43	0.084	0.42	234.77	4.03	0.95	6.17	0.15	0.47	0.52	2.90	235.17	1.32	0.31	-0.17	-0.52	2.63
11:06:29	07.92	29.71	29.23	38.38	46.14	464.50	55.75	32.99	33.32	0.083	0.42	235.42	2.79	0.66	5.45	0.12	0.50	0.56	2.72	235.46	1.32	0.31	-0.17	-0.54	2.53
11:09:44	05.90	27.73	27.32	36.34	43.51	468.05	53.10	30.92	31.08	0.081	0.42	239.52	2.81	0.67	5.51	0.12	0.44	0.50	2.45	240.70	1.24	0.30	-0.14	-0.46	2.29
11:12:59	07.34	29.17	28.68	37.73	44.89	468.96	54.43	32.51	32.72	0.083	0.42	236.48	2.89	0.68	5.51	0.12	0.45	0.51	2.50	235.99	1.31	0.31	-0.17	-0.54	2.36
11:16:34	06.86	28.75	28.26	37.19	44.18	462.84	53.80	31.97	32.19	0.082	0.42	237.45	2.63	0.63	5.00	0.13	0.48	0.55	2.42	237.77	1.29	0.31	-0.17	-0.55	2.30
11:20:00	07.29	29.17	28.69	36.85	44.03	457.99	52.72	32.22	32.46	0.083	0.41	237.10	2.33	0.55	5.22	0.11	0.48	0.54	2.50	237.97	1.29	0.31	-0.17	-0.54	2.28
11:23:29	07.06	28.98	28.47	36.55	43.60	462.36	52.64	31.98	32.19	0.082	0.41	237.57	2.40	0.57	5.15	0.11	0.48	0.54	2.45	237.67	1.29	0.31	-0.17	-0.56	2.31
11:27:01	06.77	28.72	28.22	36.11	43.36	463.04	52.24	31.70	31.79	0.082	0.41	238.33	2.38	0.57	5.09	0.11	0.49	0.55	2.50	238.09	1.28	0.30	-0.17	-0.56	2.32
11:30:26	07.16	29.12	28.64	36.55	43.73	463.76	52.72	32.18	32.25	0.083	0.41	237.47	2.42	0.57	5.23	0.11	0.48	0.54	2.49	237.75	1.29	0.31	-0.17	-0.54	2.26
11:34:17	06.47	28.45	27.97	36.10	42.97	459.53	51.13	31.33	31.40	0.081	0.41	239.12	1.85	0.44	4.67	0.09	0.50	0.57	2.35	238.80	1.27	0.30	-0.16	-0.53	2.16
11:38:30	07.80	29.80	29.26	36.96	43.70	444.08	51.06	32.76	32.91	0.083	0.40	236.45	1.59	0.38	4.24	0.09	0.55	0.63	2.35	236.45	1.30	0.31	-0.19	-0.60	2.10
11:42:43	07.50	29.52	29.00	35.99	42.28	436.71	49.54	32.45	32.53	0.083	0.39	237.14	1.50	0.36	4.26	0.08	0.51	0.59	2.19	236.43	1.30	0.31	-0.18	-0.58	1.98
11:46:33	07.39	29.44	28.87	35.78	42.05	446.59	50.54	32.46	32.45	0.083	0.39	237.27	1.81	0.43	4.68	0.09	0.47	0.53	2.18	236.17	1.31	0.31	-0.20	-0.64	2.05
11:50:44	07.37	29.46	28.91	35.69	41.77	437.95	48.60	32.40	32.41	0.083	0.39	237.47	1.18	0.28	4.28	0.07	0.49	0.57	2.11	236.07	1.32	0.31	-0.19	-0.62	2.05
11:54:58	08.36	30.47	29.90	36.42	42.65	435.33	49.32	33.32	33.43	0.084	0.39	235.64	1.21	0.28	4.24	0.07	0.52	0.59	2.19	235.71	1.32	0.31	-0.20	-0.64	1.90
11:59:23	06.21	28.35	27.85	34.23	40.10	432.59	46.69	31.11	31.03	0.081	0.38	239.96	1.18	0.28	4.07	0.07	0.49	0.57	2.01	236.09	1.31	0.31	-0.17	-0.56	1.95
12:03:37	08.17	30.30	29.74	36.01	42.10	442.18	48.47	33.29	33.32	0.083	0.38	235.92	0.69	0.16	4.23	0.04	0.51	0.58	2.14	235.85	1.32	0.31	-0.20	-0.63	1.92
12:08:10	08.07	30.23	29.70	36.09	42.19	439.45	48.27	33.23	33.26	0.083	0.38	236.15	0.64	0.15	3.94	0.04	0.54	0.62	2.14	235.94	1.32	0.31	-0.18	-0.59	1.97
12:13:00	08.03	30.20	29.66	35.76	41.54	416.63	46.83	33.13	33.18	0.083	0.38	236.43	0.49	0.12	3.71	0.03	0.55	0.63	2.02	236.51	1.31	0.31	-0.19	-0.61	1.77
12:18:02	08.04	30.25	29.71	35.27	40.89	407.59	45.66	33.09	33.15	0.083	0.37	236.56	0.25	0.06	3.56	0.02	0.55	0.64	1.97	236.79	1.30	0.31	-0.19	-0.60	1.78

E.1.2. Heat Pump Test (no engine)

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
12:48:28	02.31	15.24	15.71	18.13	16.82	013.50	18.30	26.90	27.91	0.061	0.24	248.10	1.52	0.03	9.30	0.04	-0.04	0.00	-0.34	247.54	2.96	0.73	0.12	0.32	-0.12
12:50:14	02.37	15.79	21.36	24.49	18.54	013.00	18.32	20.91	30.17	0.061	0.24	246.91	2.99	0.03	10.22	0.07	-0.15	-0.08	-1.53	246.69	3.04	0.75	1.43	1.93	-0.07
12:51:57	02.51	18.28	25.05	28.67	21.77	014.09	18.77	18.25	34.92	0.062	0.26	246.25	3.07	0.04	10.42	0.07	-0.17	-0.10	-1.79	246.52	3.10	0.77	1.75	2.32	0.21
12:53:40	02.49	20.07	28.13	31.47	24.84	016.14	19.84	17.03	39.85	0.062	0.27	246.21	3.14	0.04	10.44	0.07	-0.16	-0.09	-1.72	246.11	3.16	0.78	2.08	2.70	0.56
12:55:24	02.52	22.27	30.05	33.71	27.67	018.66	21.44	15.88	44.79	0.062	0.29	246.12	3.23	0.04	10.37	0.08	-0.15	-0.07	-1.57	246.12	3.27	0.81	2.02	2.54	0.47
12:57:07	02.48	24.49	32.63	36.77	30.54	021.41	23.25	15.28	50.81	0.062	0.31	246.16	3.33	0.04	10.44	0.08	-0.16	-0.08	-1.62	245.81	3.38	0.83	2.11	2.58	0.43
12:58:50	02.59	26.44	35.04	39.54	33.26	024.39	25.47	15.17	56.00	0.062	0.33	245.89	3.41	0.04	10.43	0.08	-0.16	-0.08	-1.64	245.73	3.40	0.84	2.24	2.67	1.09
13:00:33	02.54	24.07	34.24	38.69	34.99	027.30	27.55	14.97	59.85	0.062	0.34	246.04	3.36	0.04	10.44	0.08	-0.09	-0.01	-0.96	246.71	3.38	0.83	2.63	3.18	1.66
13:02:17	02.09	24.55	34.11	38.40	35.18	029.53	28.85	14.33	63.31	0.061	0.35	247.18	3.41	0.03	10.39	0.08	-0.08	0.00	-0.83	247.58	3.41	0.84	2.46	2.91	1.76
13:04:00	02.01	25.21	34.84	39.12	35.94	031.19	30.02	14.22	66.29	0.061	0.36	247.38	3.44	0.03	10.44	0.08	-0.08	0.00	-0.82	247.40	3.44	0.85	2.47	2.90	1.90
13:05:43	02.18	24.41	34.89	39.25	36.83	032.61	31.28	14.30	68.02	0.243	0.37	247.17	3.42	0.03	10.44	0.08	-0.24	-0.16	-2.49	246.84	3.39	0.84	10.72	12.68	18.91
13:07:26	03.94	24.79	34.63	38.91	37.01	033.63	32.40	14.51	69.88	0.506	0.37	246.83	3.43	0.03	10.40	0.08	-0.39	-0.31	-4.05	246.66	3.46	0.85	20.91	24.74	13.95
13:09:09	13.30	25.99	36.34	40.92	37.75	034.31	33.04	14.82	72.13	0.506	0.38	246.69	3.46	0.03	10.44	0.08	-0.65	-0.56	-6.74	246.65	3.43	0.85	21.98	25.74	17.61
13:10:52	22.23	25.88	36.05	40.59	38.17	035.08	33.65	14.83	72.78	0.506	0.38	246.66	3.46	0.03	10.44	0.08	-0.49	-0.41	-5.14	246.53	3.48	0.86	21.60	25.33	15.13
13:12:36	15.51	26.77	36.97	41.47	38.50	035.79	33.80	14.61	73.92	0.224	0.39	246.95	3.50	0.03	10.43	0.08	-0.27	-0.19	-2.81	246.39	3.51	0.87	9.60	11.11	15.20
13:14:19	02.25	26.72	37.31	41.83	39.25	036.50	34.36	14.68	74.78	0.506	0.39	246.86	3.49	0.03	10.41	0.08	-0.53	-0.45	-5.50	247.23	3.46	0.86	22.49	26.12	18.65
13:16:02	02.07	25.07	35.90	40.27	39.05	037.09	34.58	14.22	74.56	0.506	0.39	247.39	3.44	0.03	10.44	0.08	-0.25	-0.17	-2.60	247.40	3.45	0.85	23.00	27.03	19.66
13:17:45	02.16	25.38	35.82	40.03	38.58	052.49	34.95	14.19	75.64	0.506	0.39	247.42	3.46	0.03	10.44	0.08	-0.30	-0.21	-3.09	247.75	3.48	0.86	22.17	25.92	17.31
13:19:28	01.81	26.66	36.79	41.02	38.66	037.39	34.65	13.87	76.57	0.506	0.40	248.26	3.51	0.03	10.43	0.08	-0.48	-0.40	-5.02	248.54	3.52	0.88	21.53	24.71	16.65
13:21:11	01.84	27.31	37.63	41.83	39.44	037.73	34.87	13.86	76.05	0.416	0.40	248.28	3.52	0.03	10.44	0.08	-0.40	-0.32	-4.18	247.97	3.53	0.88	18.04	20.62	12.67
13:22:54	01.94	27.39	38.02	42.52	40.09	038.22	35.30	14.00	76.40	0.324	0.41	248.22	3.52	0.03	10.44	0.08	-0.32	-0.23	-3.32	248.19	3.50	0.87	14.47	16.55	8.17
13:24:38	02.01	27.24	37.60	41.97	40.13	038.71	35.74	14.03	76.63	0.068	0.41	248.05	3.52	0.03	10.43	0.08	-0.05	0.03	-0.53	247.72	3.55	0.88	2.96	3.39	1.83
13:26:21	02.12	28.09	38.48	43.01	40.42	039.03	36.18	14.35	77.04	0.054	0.41	247.86	3.56	0.03	10.41	0.08	-0.06	0.03	-0.59	247.88	3.55	0.88	2.35	2.67	1.77
13:28:04	02.15	28.43	38.88	43.56	40.86	039.42	36.62	14.52	77.49	0.053	0.42	247.72	3.56	0.03	10.44	0.08	-0.06	0.03	-0.61	247.85	3.56	0.88	2.34	2.65	1.81
13:29:47	02.19	28.68	39.17	43.88	41.49	039.89	37.11	14.58	78.43	0.054	0.42	247.61	3.58	0.03	10.44	0.08	-0.05	0.03	-0.54	247.54	3.58	0.89	2.36	2.66	1.91
13:31:31	02.24	28.92	39.40	44.15	42.22	040.29	37.51	14.66	79.28	0.054	0.42	247.48	3.59	0.03	10.38	0.09	-0.04	0.04	-0.44	247.34	3.61	0.89	2.37	2.67	2.00
13:33:14	02.33	29.17	39.70	44.50	42.92	040.69	37.91	14.79	79.78	0.054	0.43	247.30	3.59	0.03	10.44	0.09	-0.03	0.05	-0.36	247.32	3.60	0.89	2.40	2.70	2.08
13:34:57	02.36	29.35	39.90	44.68	43.43	041.05	38.24	14.79	79.73	0.054	0.43	247.23	3.60	0.03	10.43	0.09	-0.03	0.06	-0.28	247.17	3.59	0.89	2.40	2.70	2.19
13:36:40	02.35	28.84	39.54	44.31	43.52	041.35	38.46	14.75	80.72	0.054	0.43	247.21	3.57	0.03	10.44	0.08	-0.02	0.07	-0.18	247.10	3.58	0.89	2.43	2.76	2.14
13:38:24	02.34	29.11	39.69	44.52	43.56	041.45	38.61	14.76	80.13	0.054	0.43	247.21	3.59	0.03	10.38	0.09	-0.02	0.06	-0.22	247.18	3.58	0.89	2.41	2.72	2.24
13:40:07	02.33	29.22	39.80	44.60	43.74	041.65	38.72	14.69	80.08	0.054	0.43	247.29	3.59	0.03	10.44	0.09	-0.02	0.07	-0.20	247.20	3.60	0.89	2.42	2.73	2.25
13:41:50	02.32	29.43	39.97	44.76	44.02	041.73	38.83	14.78	80.74	0.054	0.43	247.25	3.60	0.03	10.43	0.09	-0.02	0.07	-0.17	247.13	3.60	0.89	2.41	2.71	2.24
13:43:34	02.27	29.67	40.20	45.03	44.33	041.92	38.90	14.87	81.02	0.054	0.44	247.37	3.61	0.03	10.39	0.09	-0.02	0.07	-0.16	247.62	3.62	0.90	2.41	2.70	2.31
13.45:17	02.14	29.81	40.33	45.16	44.57	042.13	38.86	14.75	80.91	0.054	0.44	247.67	3.62	0.03	10.44	0.09	-0.01	0.07	-0.14	247.75	3.61	0.89	2.40	2.68	2.30

E.1.2. Heat Pump Test (no engine)

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
13:47:00	02.06	30.14	40.57	45.40	44.87	042.38	38.92	14.68	80.56	0.054	0.44	247.86	3.63	0.03	10.44	0.09	-0.01	0.07	-0.12	247.78	3.64	0.90	2.37	2.64	2.19
13:48:43	02.09	30.80	41.12	46.03	45.36	042.59	39.15	14.74	80.67	0.054	0.45	247.81	3.66	0.03	10.44	0.09	-0.01	0.07	-0.15	247.84	3.67	0.91	2.35	2.59	2.18
13:50:26	02.06	31.46	41.62	46.68	45.83	043.00	39.33	14.69	80.58	0.054	0.45	247.87	3.68	0.03	10.40	0.09	-0.02	0.07	-0.19	247.81	3.70	0.92	2.32	2.54	2.04
13:52:10	02.01	32.25	42.18	47.44	45.91	043.47	39.59	14.73	81.23	0.054	0.46	247.95	3.71	0.03	10.44	0.09	-0.03	0.05	-0.35	248.02	3.73	0.92	2.26	2.46	1.80
13:53:53	02.00	33.29	43.06	48.27	46.39	044.07	39.93	14.82	82.18	0.054	0.47	248.01	3.75	0.03	10.44	0.09	-0.04	0.05	-0.43	248.06	3.76	0.93	2.22	2.39	1.83
13:55:36	02.02	34.59	44.34	49.18	47.04	044.74	40.40	14.94	82.40	0.054	0.48	247.95	3.78	0.03	10.42	0.09	-0.05	0.04	-0.49	247.91	3.80	0.94	2.22	2.37	1.70
13:57:19	01.98	35.65	45.30	50.14	47.83	045.60	40.91	15.08	82.93	0.054	0.49	247.99	3.82	0.03	10.43	0.09	-0.05	0.04	-0.53	248.04	3.84	0.95	2.20	2.32	1.66
13:59:02	01.99	36.53	46.04	51.04	48.62	046.44	41.49	15.18	83.43	0.054	0.50	247.96	3.87	0.03	10.44	0.09	-0.05	0.04	-0.55	248.06	3.89	0.97	2.16	2.26	1.57
14:00:45	02.02	37.52	46.81	51.94	49.49	047.36	42.13	15.32	83.87	0.054	0.51	247.96	3.90	0.03	10.44	0.09	-0.05	0.04	-0.56	248.03	3.93	0.97	2.11	2.18	1.51
14:02:29	02.03	38.58	47.68	52.91	50.41	048.30	42.79	15.60	84.55	0.054	0.52	247.88	3.95	0.03	10.42	0.09	-0.05	0.04	-0.57	247.94	3.97	0.99	2.07	2.11	1.50
14:04:12	02.06	39.69	48.56	53.89	51.36	049.31	43.53	15.73	84.35	0.054	0.53	247.84	4.00	0.03	10.43	0.10	-0.06	0.04	-0.58	247.78	4.02	1.00	2.01	2.03	1.40
14:05:55	02.10	40.71	49.49	54.89	52.33	050.34	44.29	15.91	84.89	0.054	0.55	247.75	4.05	0.03	10.39	0.10	-0.06	0.04	-0.58	247.96	4.07	1.01	1.99	1.99	1.38
14:07:39	02.01	41.56	50.27	55.81	53.15	051.36	45.00	15.97	85.74	0.054	0.56	247.79	4.10	0.03	10.38	0.10	-0.06	0.04	-0.60	247.57	4.12	1.02	1.97	1.94	1.30
14:09:23	01.66	41.72	45.16	50.32	52.39	052.38	45.39	18.92	85.11	0.054	0.55	249.18	1.21	0.02	10.39	0.03	0.04	0.07	0.47	249.13	1.09	0.27	0.77	2.57	1.14
14:11:06	01.67	39.62	39.93	44.38	48.81	051.82	45.44	22.84	82.82	0.054	0.51	249.19	1.09	0.02	10.42	0.03	0.10	0.12	1.00	249.28	1.09	0.27	0.07	0.26	0.99
14:12:49	01.63	37.65	37.78	41.92	45.98	049.89	44.27	24.49	81.04	0.054	0.48	249.31	1.09	0.02	10.44	0.03	0.09	0.11	0.92	249.32	1.09	0.27	0.03	0.11	0.93
14:14:32	01.62	36.64	36.58	40.55	43.62	047.52	42.74	25.07	79.30	0.054	0.45	249.42	1.09	0.02	10.43	0.03	0.07	0.09	0.70	249.41	1.09	0.27	-0.01	-0.05	0.40
14:16:15	01.54	35.99	35.85	39.72	40.56	045.24	41.14	25.33	77.54	0.054	0.43	249.61	1.09	0.02	10.44	0.03	0.02	0.04	0.19	249.71	1.09	0.27	-0.03	-0.11	0.05
14:17:58	01.43	35.53	35.36	39.15	39.39	043.22	39.70	25.41	75.82	0.054	0.42	249.87	1.09	0.02	10.44	0.03	0.01	0.03	0.05	249.92	1.09	0.27	-0.04	-0.14	-0.02
14:19:41	01.38	34.95	34.81	38.51	38.64	041.54	38.50	25.51	74.30	0.054	0.41	250.00	1.08	0.02	10.44	0.03	0.00	0.03	0.03	250.08	1.09	0.27	-0.03	-0.12	0.00
14:21:24	01.36	34.21	34.14	37.73	37.86	040.19	37.54	25.60	72.72	0.054	0.40	250.07	1.08	0.02	10.44	0.03	0.00	0.03	0.03	250.08	1.08	0.27	-0.02	-0.06	0.04
14:23:08	01.31	33.32	33.49	36.96	37.11	039.08	36.66	25.63	71.59	0.054	0.39	250.20	1.08	0.02	10.44	0.03	0.00	0.03	0.04	250.17	1.08	0.27	0.04	0.14	0.10

E.1.3. HP Mode Test

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
14:58:41	26.01	24.79	31.84	34.34	33.54	391.54	43.70	18.03	25.89	0.059	0.05	239.44	3.43	0.93	7.30	0.11	-0.03	0.09	-0.20	239.45	3.48	0.83	1.76	2.14	2.05
15:01:11	29.56	28.38	35.99	38.83	42.65	422.49	52.36	16.26	26.03	0.053	0.10	239.47	3.57	0.94	7.14	0.12	0.12	0.24	0.85	239.48	3.65	0.87	1.69	1.98	2.72
15:03:43	32.73	31.30	39.52	42.85	49.49	425.18	57.32	16.62	26.13	0.069	0.16	239.43	3.68	0.92	7.08	0.12	0.27	0.40	1.92	239.35	3.71	0.89	2.38	2.70	4.38
15:06:16	34.84	33.61	42.05	45.76	53.04	424.48	60.40	17.13	26.11	0.068	0.20	239.50	3.78	0.92	7.04	0.13	0.30	0.43	2.09	239.47	3.81	0.91	2.42	2.67	4.53
15:08:49	36.00	35.39	43.59	47.66	55.66	424.87	62.23	17.54	25.35	0.068	0.23	239.53	3.84	0.91	7.02	0.13	0.33	0.46	2.29	239.62	3.83	0.92	2.34	2.54	4.77
15:11:22	39.38	38.37	46.70	50.24	57.74	425.91	63.58	18.01	25.63	0.068	0.26	239.56	3.96	0.92	7.04	0.13	0.30	0.44	2.14	239.59	4.00	0.96	2.37	2.49	4.61
15:13:54	41.93	41.07	45.48	49.25	59.96	428.80	64.71	19.58	26.21	0.063	0.29	240.74	1.87	0.93	7.05	0.06	0.40	0.46	2.83	241.14	1.22	0.29	1.16	2.59	3.73
0:00:00	38.39	38.37	41.38	44.36	57.53	428.58	62.82	24.83	27.82	0.062	0.26	240.29	2.75	0.93	7.05	0.09	0.48	0.58	3.41	239.67	3.96	0.95	0.78	1.18	4.34
15:19:00	36.96	36.56	45.98	49.30	58.37	427.62	64.01	17.97	25.20	0.062	0.27	239.67	3.79	0.93	7.04	0.13	0.33	0.46	2.36	239.65	3.78	0.90	2.44	2.68	4.90
15:21:32	37.00	36.59	45.83	49.35	59.27	426.74	64.53	17.08	24.50	0.062	0.28	239.67	3.80	0.93	7.06	0.13	0.36	0.49	2.57	239.67	3.81	0.91	2.39	2.63	4.99
15:24:05	38.77	38.01	44.79	48.61	59.67	426.82	64.80	18.33	24.72	0.061	0.29	240.34	2.52	0.93	7.06	0.09	0.40	0.49	2.85	240.99	1.29	0.31	1.74	2.88	4.60
15:26:38	37.47	37.15	41.40	44.62	56.77	423.48	62.53	25.22	27.80	0.061	0.26	240.05	3.21	0.93	7.04	0.11	0.44	0.55	3.13	239.75	3.72	0.89	1.09	1.42	4.13
15:29:11	37.66	37.00	45.92	49.65	58.18	422.01	63.98	17.47	24.70	0.061	0.28	239.74	3.74	0.92	7.03	0.13	0.31	0.44	2.19	239.81	3.74	0.90	2.28	2.55	4.56
15:31:45	38.05	37.53	46.66	50.46	59.52	418.41	64.67	16.98	24.21	0.061	0.29	239.85	3.77	0.91	6.96	0.13	0.33	0.46	2.32	239.93	3.75	0.90	2.34	2.59	4.78
15:34:19	37.76	37.20	46.44	50.09	60.02	416.98	64.96	16.87	24.11	0.061	0.29	239.89	3.76	0.91	7.00	0.13	0.36	0.49	2.55	239.88	3.75	0.90	2.36	2.62	4.90
15:36:53	**37.95**	**37.40**	**46.67**	**50.33**	**60.17**	**415.67**	**64.98**	**16.74**	**23.99**	**0.061**	**0.29**	**239.90**	**3.77**	**0.90**	**7.00**	**0.13**	**0.36**	**0.49**	**2.52**	**239.96**	**3.77**	**0.90**	**2.37**	**2.62**	**4.92**
15:39:28	38.03	37.47	46.85	50.62	60.29	414.46	65.07	17.00	24.42	0.061	0.29	239.82	3.86	0.89	6.96	0.13	0.36	0.49	2.48	239.82	3.93	0.94	2.40	2.59	4.89
15:42:03	38.12	37.48	47.07	50.98	60.47	413.82	65.28	18.39	25.61	0.061	0.29	239.77	3.92	0.89	6.94	0.14	0.35	0.49	2.44	239.75	3.91	0.94	2.46	2.62	4.94
15:44:38	38.16	37.52	47.19	51.06	60.48	409.75	65.29	18.67	25.76	0.061	0.30	239.77	3.94	0.89	6.92	0.14	0.35	0.49	2.43	239.72	3.94	0.95	2.49	2.64	4.95
15:47:12	38.01	37.37	43.78	47.57	60.32	413.19	65.09	21.07	26.36	0.061	0.29	240.67	2.33	0.89	6.98	0.08	0.47	0.55	3.26	241.46	1.25	0.30	1.63	2.92	4.96

E.1.4. CHPHP Mode Test - 1

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
13:02:02	27.98	18.13	19.44	20.31	24.43	427.63	38.67	26.56	27.98	0.058	0.18	241.76	4.80	1.16	7.16	0.05	0.14	0.19	1.00	242.96	1.25	0.30	0.32	0.84	1.41
13:04:36	28.35	19.68	20.91	22.38	28.79	476.45	43.27	24.82	28.35	0.057	0.21	242.90	4.77	1.16	6.99	0.06	0.22	0.28	1.54	242.76	1.23	0.30	0.30	0.75	1.90
13:07:11	28.53	21.63	21.90	23.42	32.02	483.90	45.49	27.28	28.53	0.057	0.23	243.04	4.74	1.15	6.92	0.05	0.30	0.34	2.06	243.12	1.25	0.30	0.06	0.21	2.07
13:09:47	29.53	23.79	24.49	26.26	34.42	485.50	47.14	27.07	29.53	0.057	0.25	242.42	4.74	1.15	6.90	0.07	0.29	0.36	1.97	242.52	1.25	0.30	0.17	0.34	2.19
13:12:24	32.08	25.36	25.24	26.72	36.85	487.23	49.28	28.19	32.08	0.058	0.26	242.37	4.74	1.15	6.87	0.05	0.36	0.41	2.46	241.19	3.07	0.74	-0.03	-0.09	2.43
13:15:01	34.53	26.44	27.36	29.28	39.03	487.41	50.75	26.59	34.53	0.058	0.28	242.26	4.72	1.14	6.86	0.04	0.35	0.39	2.37	242.42	1.19	0.29	0.22	0.77	2.63
13:17:38	35.66	25.74	27.79	30.75	38.83	486.39	49.98	24.90	35.66	0.057	0.29	242.96	4.69	1.14	6.84	0.07	0.28	0.35	1.93	243.45	1.17	0.28	0.49	1.02	2.56
13:20:16	36.74	25.80	27.97	30.82	39.23	486.83	50.26	24.07	36.74	0.057	0.30	243.03	4.69	1.14	6.82	0.07	0.29	0.37	2.01	243.55	1.16	0.28	0.52	1.05	2.33
13:22:54	37.29	26.56	28.80	31.87	39.39	486.80	50.16	23.48	37.29	0.056	0.31	244.31	4.65	1.14	6.80	0.09	0.26	0.35	1.78	244.75	1.19	0.29	0.53	0.89	2.40
13:25:32	38.93	27.45	28.84	31.33	40.33	486.46	51.10	24.30	38.93	0.056	0.31	244.65	4.65	1.14	6.79	0.07	0.31	0.39	2.12	243.72	3.35	0.82	0.33	0.66	2.38
13:28:10	41.43	26.79	28.22	31.03	42.36	486.65	52.24	22.87	41.43	0.056	0.32	244.93	4.65	1.14	6.81	0.04	0.39	0.43	2.66	244.94	1.17	0.29	0.33	1.16	2.59
13:30:49	43.46	27.86	32.33	35.18	41.84	486.44	51.93	21.06	43.46	0.056	0.32	243.67	4.67	1.14	6.78	0.12	0.23	0.35	1.58	242.95	3.44	0.84	1.06	1.32	2.98
13:33:28	50.44	29.78	37.33	41.21	46.88	486.85	55.40	16.51	50.44	0.056	0.36	242.90	4.68	1.14	6.76	0.13	0.20	0.33	1.35	242.88	3.51	0.85	1.79	2.10	3.42
13:36:08	57.46	30.28	38.30	41.97	49.03	487.21	57.05	15.25	57.46	0.056	0.38	243.28	4.68	1.14	6.75	0.13	0.25	0.38	1.67	243.61	3.60	0.88	1.89	2.18	3.49
13:38:48	59.82	29.46	38.36	44.60	48.48	494.08	55.80	11.52	59.82	0.049	0.42	249.87	4.57	1.14	6.72	0.13	0.12	0.25	0.79	250.33	3.58	0.90	1.81	2.03	-1.28
13:41:28	65.63	31.88	39.61	181.82	50.02	494.61	57.01	11.83	65.63	0.060	0.44	249.38	4.59	1.14	6.71	0.14	-4.92	-4.79	-33.03	248.72	3.66	0.91	1.93	2.13	-78.20
13:44:09	70.12	32.79	40.92	93.85	51.12	494.28	57.93	11.88	70.12	0.060	0.46	249.28	4.57	1.14	6.70	0.14	-1.60	-1.47	-10.74	249.23	3.66	0.91	2.04	2.23	3.78
13:46:49	73.07	32.07	40.61	44.00	51.83	494.69	58.36	11.93	73.07	0.060	0.46	249.18	4.58	1.14	6.70	0.14	0.29	0.43	1.97	248.96	3.69	0.92	2.14	2.35	4.00
13:49:30	75.73	33.32	41.94	45.01	52.51	494.54	58.92	12.23	75.73	0.060	0.47	248.89	4.58	1.14	6.69	0.14	0.28	0.42	1.89	248.82	3.71	0.92	2.17	2.35	4.18
13:52:11	76.51	33.99	42.80	45.75	53.24	493.69	59.38	12.28	76.51	0.060	0.48	248.87	4.58	1.14	6.69	0.14	0.28	0.42	1.89	248.93	3.73	0.93	2.22	2.39	4.15
13:54:52	78.22	32.44	41.25	44.37	53.09	495.03	59.33	12.17	78.22	0.060	0.48	248.94	4.58	1.14	6.69	0.14	0.33	0.47	2.20	248.91	3.71	0.92	2.22	2.42	4.14
13:57:33	78.41	33.63	42.65	45.60	53.11	494.92	59.34	12.16	78.41	0.060	0.48	248.98	4.57	1.14	6.68	0.14	0.28	0.42	1.89	249.00	3.73	0.93	2.27	2.44	4.21
14:00:14	78.15	33.75	42.62	45.43	53.39	495.42	59.59	12.19	78.15	0.060	0.48	248.86	4.59	1.14	6.70	0.14	0.30	0.44	2.01	248.77	3.73	0.93	2.23	2.40	4.20
14:02:55	**79.14**	**31.86**	**41.22**	**44.12**	**53.17**	**495.80**	**59.51**	**17.00**	**24.10**	**0.060**	**0.48**	**248.92**	**4.58**	**1.14**	**6.65**	**0.14**	**0.34**	**0.48**	**2.28**	**248.96**	**3.69**	**0.92**	**2.35**	**2.58**	**4.45**
14:05:36	77.73	32.50	41.45	44.33	52.81	495.97	59.19	12.06	77.73	0.060	0.48	248.95	4.58	1.14	6.70	0.14	0.32	0.46	2.13	248.95	3.70	0.92	2.25	2.45	4.38
14:08:16	77.28	30.72	40.05	42.56	52.30	496.47	58.88	11.89	77.28	0.060	0.47	248.96	4.58	1.14	6.71	0.13	0.36	0.50	2.45	249.17	3.68	0.92	2.34	2.58	4.27
14:10:57	75.28	30.57	39.78	42.81	51.33	495.65	58.03	11.58	75.28	0.059	0.47	249.42	4.57	1.14	6.70	0.14	0.32	0.45	2.13	249.31	3.69	0.92	2.29	2.52	4.01
14:13:38	74.57	32.61	41.38	44.42	51.95	495.29	58.46	11.83	74.57	0.060	0.47	249.26	4.57	1.14	6.69	0.14	0.28	0.42	1.89	249.47	3.69	0.92	2.19	2.38	4.06
14:16:19	74.49	32.48	34.26	37.19	50.14	495.24	57.01	17.00	74.49	0.059	0.44	250.53	4.55	1.14	6.69	0.04	0.48	0.52	3.22	250.41	1.07	0.27	0.44	1.63	3.22
14:19:00	71.54	30.86	36.77	39.52	46.89	494.45	54.67	15.99	71.54	0.059	0.41	249.62	4.56	1.14	6.69	0.13	0.28	0.41	1.84	249.66	3.68	0.92	1.47	1.66	3.41
14:21:41	70.74	32.99	41.32	43.89	50.38	494.42	57.53	11.88	70.74	0.060	0.44	249.16	4.57	1.14	6.69	0.14	0.24	0.38	1.63	249.17	3.74	0.93	2.09	2.25	3.81
14:27:33	71.47	35.19	41.27	43.63	52.04	481.19	57.51	13.23	71.47	0.060	0.46	250.60	2.73	0.68	3.06	0.19	0.69	0.88	2.11	252.39	1.06	0.27	1.52	2.69	3.18

E.1.5. CHP Test - 2

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
13:46:14	05.24	25.62	25.66	24.92	23.29	016.12	23.38	29.25	29.31	0.076	0.39	240.89	1.24	0.00	10.19	0.03	-0.05	-0.02	-0.52	241.08	1.24	0.30	0.01	0.04	-0.54
13:48:00	05.03	25.41	25.44	24.78	23.03	016.24	23.14	29.08	29.11	0.076	0.39	241.29	1.23	0.00	10.18	0.03	-0.05	-0.03	-0.55	241.25	1.23	0.30	0.01	0.04	-0.52
13:49:45	04.98	25.40	25.42	24.72	22.97	016.21	23.12	29.08	29.14	0.076	0.39	241.33	1.23	0.00	10.21	0.03	-0.05	-0.03	-0.56	241.49	1.23	0.30	0.01	0.03	-0.52
13:51:31	05.00	25.40	25.44	24.66	22.95	016.20	23.11	29.07	29.12	0.076	0.39	241.30	1.23	0.00	10.18	0.03	-0.05	-0.02	-0.55	241.31	1.23	0.30	0.01	0.04	-0.54
13:53:16	05.68	26.09	25.91	24.66	23.61	016.18	23.83	29.78	29.88	0.076	0.39	239.88	1.25	0.00	10.24	0.03	-0.03	0.00	-0.34	239.54	1.25	0.30	-0.06	-0.19	-0.40
13:55:01	06.23	26.62	26.38	25.27	24.43	133.87	25.83	30.61	30.78	0.077	0.39	238.26	1.28	0.00	10.25	0.03	-0.03	0.00	-0.27	238.26	1.28	0.31	-0.08	-0.26	-0.38
13:56:46	05.85	26.29	26.07	26.24	25.66	327.40	29.01	30.54	30.71	0.077	0.40	238.41	1.30	0.00	10.25	0.03	-0.02	0.01	-0.19	239.63	1.26	0.30	-0.07	-0.23	-0.27
13:58:31	05.20	25.63	25.49	27.24	26.98	386.30	30.39	30.07	30.20	0.076	0.41	239.39	1.27	0.00	10.22	0.03	-0.01	0.02	-0.08	239.45	1.27	0.30	-0.05	-0.15	0.07
14:00:16	05.10	25.55	25.40	28.51	28.92	407.87	32.20	30.03	30.17	0.076	0.43	239.46	1.26	0.00	10.21	0.03	0.01	0.04	0.13	239.74	1.27	0.30	-0.05	-0.15	0.08
14:02:02	05.02	25.50	25.35	29.73	30.41	416.76	33.78	30.02	30.15	0.076	0.44	239.57	1.28	0.00	10.23	0.03	0.02	0.05	0.22	239.60	1.27	0.30	-0.05	-0.15	0.25
14:03:47	06.95	27.42	27.24	32.32	34.78	476.87	46.72	32.10	32.33	0.078	0.46	234.16	1.35	0.97	10.21	0.03	0.08	0.11	0.81	234.85	1.34	0.32	-0.06	-0.19	0.92
14:05:32	06.47	26.96	26.81	34.29	37.70	501.35	51.27	31.95	32.19	0.078	0.49	234.56	1.35	1.00	10.25	0.03	0.11	0.14	1.12	234.71	1.35	0.32	-0.05	-0.15	1.15
14:07:17	05.96	26.45	26.34	35.92	39.83	504.55	52.93	31.73	31.96	0.077	0.51	235.09	1.35	0.98	10.25	0.03	0.12	0.15	1.27	235.10	1.34	0.32	-0.04	-0.12	1.37
14:09:02	06.15	26.67	26.52	37.08	42.00	506.53	54.89	31.84	32.07	0.077	0.54	234.88	1.35	0.98	10.22	0.03	0.16	0.19	1.60	234.53	1.38	0.32	-0.05	-0.15	1.73
14:10:48	06.12	26.66	26.50	37.32	43.25	508.39	55.85	31.73	31.98	0.077	0.55	235.07	1.34	0.98	10.21	0.03	0.19	0.22	1.93	234.66	1.35	0.32	-0.05	-0.16	1.94
14:12:33	06.65	27.18	27.03	38.15	44.38	508.39	57.02	32.53	32.81	0.078	0.56	233.56	1.36	0.98	10.24	0.03	0.20	0.23	2.04	232.16	1.38	0.32	-0.05	-0.15	1.89
14:14:19	06.87	27.38	27.24	38.87	45.00	507.87	57.68	33.21	33.52	0.078	0.56	232.20	1.39	0.98	10.10	0.03	0.20	0.23	2.02	232.38	1.37	0.32	-0.05	-0.14	2.01
14:16:05	07.16	27.71	27.56	38.96	45.55	507.97	58.20	33.54	33.88	0.079	0.56	231.62	1.39	0.98	10.22	0.03	0.21	0.24	2.18	230.56	1.41	0.32	-0.05	-0.15	2.31
14:17:50	07.11	27.69	27.54	38.49	45.63	507.20	58.19	33.42	33.75	0.079	0.55	231.87	1.40	0.98	10.21	0.03	0.23	0.26	2.36	232.09	1.39	0.32	-0.05	-0.15	2.29
14:19:35	07.78	28.38	28.20	38.94	46.43	506.67	58.96	33.95	34.28	0.079	0.53	230.82	1.41	0.98	10.22	0.03	0.24	0.28	2.49	231.18	1.40	0.32	-0.06	-0.18	2.51
14:21:21	08.10	28.68	28.50	39.04	46.72	506.17	59.33	34.07	34.38	0.079	0.52	230.63	1.41	0.98	10.19	0.03	0.25	0.28	2.57	229.99	1.45	0.33	-0.06	-0.18	2.60
14:23:06	08.38	29.02	28.79	39.54	47.22	507.13	59.77	34.50	34.82	0.080	0.51	229.96	1.42	0.98	10.25	0.03	0.25	0.28	2.58	230.36	1.41	0.32	-0.08	-0.23	2.45
14:24:51	07.84	28.48	28.29	39.76	46.74	507.66	59.40	34.01	34.34	0.079	0.52	231.06	1.40	0.98	10.24	0.03	0.23	0.26	2.32	232.72	1.37	0.32	-0.06	-0.19	2.08
14:26:36	06.96	27.61	27.47	39.18	46.09	510.57	58.86	32.86	33.17	0.078	0.52	233.26	1.38	0.99	10.23	0.03	0.22	0.25	2.26	233.01	1.37	0.32	-0.05	-0.15	2.33
14:28:21	06.81	27.50	27.37	39.46	46.87	507.51	58.84	32.62	32.93	0.078	0.52	233.73	1.36	0.97	10.23	0.03	0.24	0.27	2.42	234.87	1.34	0.32	-0.04	-0.14	2.33
14:30:06	06.36	27.10	26.96	39.17	46.81	505.91	58.43	31.95	32.26	0.077	0.53	234.97	1.35	0.96	10.27	0.03	0.24	0.27	2.47	235.09	1.34	0.32	-0.05	-0.15	2.53
14:31:51	06.31	27.06	26.92	38.96	47.08	506.13	58.46	31.87	32.16	0.077	0.53	235.14	1.35	0.96	10.26	0.03	0.26	0.29	2.62	235.06	1.34	0.32	-0.04	-0.14	2.54
14:33:36	06.29	27.04	26.92	38.57	46.82	507.05	58.40	31.81	32.07	0.077	0.53	235.24	1.34	0.96	10.20	0.03	0.26	0.29	2.67	235.61	1.37	0.32	-0.04	-0.13	2.72
14:35:22	06.86	27.63	27.47	39.64	47.34	507.05	58.85	32.28	32.60	0.077	0.52	234.31	1.36	0.96	10.22	0.03	0.25	0.28	2.51	233.37	1.37	0.32	-0.05	-0.16	2.99
14:37:06	06.93	27.72	27.55	38.33	46.82	486.36	56.09	31.82	32.12	0.077	0.52	236.09	1.34	0.20	10.28	0.03	0.27	0.30	2.76	237.84	1.30	0.31	-0.05	-0.17	2.36
14:38:51	06.45	27.26	27.12	37.55	43.63	288.95	48.74	31.01	31.25	0.077	0.50	238.29	1.29	0.00	10.28	0.03	0.19	0.22	1.96	238.63	1.29	0.31	-0.05	-0.15	1.58
14:40:36	06.46	27.27	27.13	35.79	40.30	142.00	45.04	31.04	31.21	0.077	0.48	238.29	1.29	0.00	10.25	0.03	0.14	0.17	1.46	238.37	1.29	0.31	-0.05	-0.15	1.25
14:42:21	06.50	27.34	27.18	34.17	37.75	085.21	41.98	31.09	31.23	0.077	0.46	238.18	1.29	0.00	10.27	0.03	0.11	0.14	1.16	238.24	1.29	0.31	-0.05	-0.16	1.01
14:44:05	06.47	27.29	27.14	32.61	35.68	059.06	39.24	31.02	31.13	0.077	0.44	238.24	1.29	0.00	10.28	0.03	0.10	0.13	1.00	238.08	1.30	0.31	-0.05	-0.15	0.84
14:45:50	06.76	27.59	27.43	31.61	34.33	045.26	37.64	31.26	31.41	0.077	0.43	237.69	1.30	0.00	10.28	0.03	0.09	0.12	0.88	237.52	1.30	0.31	-0.05	-0.17	0.81

E.1.5. CHP Test - 2

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
14:47:35	06.84	27.69	27.51	30.69	32.91	037.36	35.90	31.22	31.40	0.077	0.41	237.62	1.30	0.00	10.28	0.03	0.07	0.10	0.72	237.53	1.30	0.31	-0.06	-0.19	0.68
14:49:20	06.76	27.58	27.44	29.71	31.55	032.28	34.24	31.14	31.31	0.077	0.40	237.68	1.30	0.00	10.28	0.03	0.06	0.09	0.60	237.83	1.30	0.31	-0.05	-0.15	0.44
14:51:04	06.76	27.60	27.44	28.97	30.53	028.81	32.96	31.13	31.33	0.077	0.40	237.69	1.30	0.00	10.28	0.03	0.05	0.08	0.51	237.83	1.30	0.31	-0.05	-0.17	0.43
14:52:49	06.76	27.61	27.44	28.31	29.51	026.31	31.86	31.10	31.34	0.077	0.39	237.73	1.30	0.00	10.27	0.03	0.04	0.07	0.39	237.70	1.30	0.31	-0.06	-0.18	0.28
14:54:34	06.77	27.62	27.43	27.71	28.52	024.38	30.89	31.09	31.31	0.077	0.38	237.72	1.30	0.00	10.28	0.03	0.03	0.06	0.26	238.02	1.30	0.31	-0.06	-0.20	0.17
14:56:18	06.66	27.50	27.34	27.09	27.72	022.90	29.90	30.96	31.21	0.077	0.38	237.92	1.30	0.00	10.28	0.03	0.02	0.05	0.20	238.57	1.29	0.31	-0.05	-0.17	0.01
14:58:03	06.28	27.12	26.99	26.55	26.80	021.66	28.78	30.74	30.96	0.077	0.38	238.37	1.29	0.00	10.27	0.03	0.01	0.04	0.08	238.37	1.29	0.31	-0.04	-0.14	0.05
14:59:48	06.39	27.23	27.08	26.30	26.53	020.65	28.26	30.84	31.07	0.077	0.37	238.27	1.29	0.00	10.28	0.03	0.01	0.04	0.08	238.08	1.30	0.31	-0.05	-0.16	0.05
15:01:32	06.51	27.34	27.19	26.06	26.32	019.84	27.78	30.92	31.14	0.077	0.37	238.12	1.30	0.00	10.27	0.03	0.01	0.04	0.08	238.31	1.29	0.31	-0.05	-0.16	-0.01
15:03:17	06.41	27.23	27.09	25.89	25.94	019.13	27.14	30.84	31.04	0.077	0.37	238.21	1.29	0.00	10.26	0.03	0.00	0.03	0.01	238.15	1.29	0.31	-0.04	-0.14	-0.14

E.1.6. CHPHP Test - 2

Time	T1	T2	T3	T4	T5	T6	T7	T8	T9	m - Wt	Pwt	Vgen	Igen	we	Qfuel	Eff - W	Eff - th	Eff - tot	Qehe	V-hp	I-hp	whp	Qhp	COP	QT
h-m-s	C	C	C	C	C	C	C	C	C	Lt/s	Bar	V	Amp	kW	kW	%	%	%	kW	V	A	kW	kW		kW
14:33:02	25.95	25.07	25.90	27.65	31.66	382.28	45.24	28.48	31.13	0.063	0.05	237.87	2.40	0.90	8.64	0.07	0.12	0.19	1.06	237.44	3.28	0.78	0.22	0.38	1.62
14:35:30	29.75	28.54	32.94	35.61	38.83	409.72	50.54	17.38	28.21	0.063	0.08	237.44	3.39	0.93	7.31	0.11	0.12	0.23	0.85	237.23	3.45	0.82	1.16	1.43	2.30
14:38:01	33.94	32.48	38.89	41.84	46.10	412.13	55.72	17.96	27.26	0.063	0.12	237.27	3.59	0.91	7.10	0.12	0.16	0.28	1.12	237.28	3.64	0.86	1.69	1.98	3.30
14:40:34	35.55	34.21	42.11	45.46	51.79	412.56	59.71	18.61	27.08	0.062	0.16	237.29	3.72	0.90	7.05	0.13	0.24	0.36	1.66	237.27	3.74	0.89	2.07	2.34	3.91
14:43:07	36.82	35.79	44.09	47.79	55.01	412.89	62.03	19.26	27.19	0.062	0.20	237.23	3.82	0.90	7.04	0.13	0.27	0.40	1.89	237.16	3.84	0.91	2.17	2.39	4.11
14:45:39	38.06	37.36	45.93	49.85	57.65	414.34	63.80	20.07	27.71	0.062	0.23	236.58	3.88	0.90	7.04	0.13	0.29	0.42	2.04	236.21	3.89	0.92	2.24	2.44	4.35
14:48:13	38.57	37.92	46.93	51.12	59.31	414.15	64.89	20.83	28.02	0.063	0.24	236.14	3.92	0.90	7.03	0.13	0.31	0.44	2.16	236.17	3.94	0.93	2.37	2.56	4.64
14:50:45	38.92	38.26	47.30	51.13	60.27	414.89	65.24	19.00	26.96	0.062	0.26	237.02	3.81	0.90	7.08	0.13	0.34	0.47	2.40	237.08	3.80	0.90	2.37	2.62	4.82
14:53:17	39.14	38.55	47.74	51.56	60.93	415.25	65.62	18.33	26.42	0.062	0.27	237.06	3.83	0.90	7.05	0.13	0.35	0.48	2.46	237.08	3.83	0.91	2.40	2.65	4.93
14:55:50	39.38	38.81	48.12	52.03	61.58	415.19	66.19	18.64	26.56	0.062	0.27	236.48	3.83	0.90	7.03	0.13	0.36	0.49	2.51	236.61	3.82	0.90	2.44	2.70	4.99
14:58:23	39.83	39.11	48.35	52.31	61.69	414.81	66.30	18.60	26.65	0.062	0.28	236.65	3.85	0.90	7.03	0.13	0.35	0.48	2.46	236.77	3.85	0.91	2.42	2.65	4.84
15:00:56	41.99	41.03	49.95	53.97	62.70	414.86	66.91	18.96	26.97	0.062	0.29	236.80	3.93	0.90	7.01	0.13	0.33	0.46	2.29	236.76	3.94	0.93	2.33	2.50	4.42
15:03:30	42.18	41.86	42.85	46.21	61.49	414.88	65.70	22.54	28.18	0.062	0.28	238.24	1.33	0.89	6.99	0.05	0.57	0.61	3.97	238.15	1.32	0.31	0.26	0.82	3.98
15:06:04	40.18	39.94	40.44	43.51	57.15	414.36	62.76	27.67	29.69	0.062	0.25	237.95	1.32	0.89	6.99	0.04	0.51	0.56	3.57	237.84	1.30	0.31	0.13	0.41	3.58
15:11:13	38.60	38.27	38.64	41.46	53.78	413.15	60.91	29.93	31.02	0.063	0.22	237.11	1.32	0.89	3.49	0.09	0.93	1.02	3.25	236.81	1.31	0.31	0.10	0.31	3.29
15:13:47	38.11	37.78	38.11	40.97	52.55	412.41	59.95	30.69	31.57	0.063	0.20	236.62	1.33	0.89	6.96	0.05	0.44	0.49	3.08	236.63	1.32	0.31	0.09	0.28	3.13
15:16:22	37.81	37.42	37.62	40.70	52.00	412.47	59.34	31.04	31.67	0.063	0.19	236.58	1.33	0.89	6.97	0.05	0.43	0.48	3.01	236.57	1.32	0.31	0.05	0.17	3.06
15:18:56	37.53	37.12	37.41	40.48	51.72	411.98	59.10	31.28	31.80	0.063	0.19	236.50	1.33	0.89	6.98	0.05	0.43	0.47	3.00	236.51	1.32	0.31	0.08	0.25	3.11
15:21:30	36.60	35.92	36.48	39.48	51.34	411.78	58.76	31.52	31.91	0.063	0.18	236.53	1.33	0.89	6.99	0.05	0.45	0.50	3.16	236.48	1.32	0.31	0.15	0.47	3.33
15:24:04	35.96	35.20	35.87	38.73	50.93	411.07	58.26	31.63	32.02	0.063	0.18	236.46	1.33	0.89	6.97	0.05	0.47	0.51	3.25	236.34	1.32	0.31	0.18	0.57	3.47
15:26:38	35.34	34.44	35.13	37.81	50.64	410.73	57.69	31.67	31.89	0.063	0.17	236.74	1.33	0.89	6.97	0.05	0.49	0.53	3.41	236.77	1.32	0.31	0.18	0.58	3.56
15:29:12	35.31	34.35	34.97	37.61	50.02	410.60	57.37	31.81	31.90	0.063	0.17	236.74	1.33	0.89	6.99	0.04	0.47	0.52	3.30	236.68	1.32	0.31	0.17	0.53	3.38
15:31:46	35.33	34.38	34.95	37.66	49.89	410.38	57.23	31.85	31.97	0.063	0.17	236.65	1.33	0.90	6.99	0.05	0.47	0.51	3.26	236.73	1.35	0.32	0.15	0.48	3.37
15:34:20	35.37	34.47	35.07	37.82	49.74	409.59	57.14	31.99	31.99	0.063	0.16	236.72	1.36	0.89	6.98	0.05	0.45	0.50	3.17	236.58	1.36	0.32	0.16	0.49	3.32
15:37:49	35.51	34.63	35.20	38.00	49.82	408.83	57.25	32.28	32.05	0.063	0.16	236.70	1.36	0.85	5.14	0.06	0.61	0.67	3.14	237.94	1.34	0.32	0.15	0.47	3.28
15:42:58	33.31	32.64	33.50	35.83	43.94	322.34	46.83	30.93	30.52	0.062	0.13	241.00	1.31	0.00	3.49	0.09	0.61	0.70	2.12	241.07	1.29	0.31	0.22	0.71	2.29
15:48:04	31.19	30.70	30.89	33.27	40.23	312.50	43.22	31.00	30.62	0.062	0.10	240.79	1.31	0.00	3.52	0.09	0.52	0.61	1.83	240.77	1.29	0.31	0.05	0.16	1.79
15:52:52	30.28	29.77	29.82	32.11	38.35	310.32	40.93	31.00	30.75	0.063	0.08	240.41	1.32	0.00	3.74	0.08	0.44	0.52	1.64	240.41	1.30	0.31	0.01	0.05	1.64
15:55:22	33.20	31.94	39.73	42.99	43.64	390.97	53.18	20.92	28.35	0.064	0.11	234.98	3.58	0.92	7.16	0.12	0.02	0.14	0.18	235.04	3.60	0.85	2.10	2.50	2.50
15:57:56	36.88	35.47	43.41	47.01	51.79	409.55	59.59	17.48	26.97	0.064	0.16	235.10	3.71	0.90	7.00	0.12	0.18	0.31	1.29	235.08	3.73	0.88	2.13	2.44	3.52

Appendix E.2 Derivation of Sensitivity Analysis

This appendix derives the sensitivity analysis used in Chapter 6. Derived sensitivity functions will be employed with appropriate transducer tolerances in Appendix E.3 to calculate tolerance of analytical results.

E.2.1 Generator Power Output

$$w_e = I_{gen} V_{gen} \tag{E.1}$$

Note V_{gen} represents the RMS voltage of the generator.

Applying sensitivity analysis to Equation *E.1* gives:

$$\Delta w_e = \sqrt{\left[\frac{d(I_{gen}V_{gen})}{dI_{gen}}\Delta I_{gen}\right]^2 + \left[\frac{d(I_{gen}V_{gen})}{dV_{gen}}\Delta V_{gen}\right]^2} \tag{E.2}$$

Differentiating gives:

$$\Delta w_e = \sqrt{(V_{gen}\Delta I_{gen})^2 + (I_{gen}\Delta V_{gen})^2} \tag{E.3}$$

E.2.2 Engine/Generator Set Electrical Conversion Efficiency

$$\text{From: } \eta_e = \frac{w_e}{Q_f} \tag{E.4}$$

$$\Delta \eta_e = \sqrt{\left[\frac{d(w_e/Q_f)}{dQ_f}\Delta Q_f\right]^2 + \left[\frac{d(w_e/Q_f)}{dw_e}\Delta w_e\right]^2} \tag{E.5}$$

$$\Delta \eta_e = \sqrt{\left(\frac{-w_e}{Q_f^{\,2}}\Delta Q_f\right)^2 + \left(\frac{1}{Q_f}\Delta w_e\right)^2} \tag{E.6}$$

E.2.3 EHE Thermal Output

From: $Q_{ehe} = \dot{m}_{wt} Cp_{wt} (T_5 - T_4)$ *(E.7)*

$$\Delta Q_{ehe} = \sqrt{\begin{aligned}&\left\{\frac{d[\dot{m}_{wt} Cp_{wt}(T_5 - T_4)]}{d\dot{m}_{wt}}\Delta \dot{m}_{wt}\right\}^2 + \left\{\frac{d[\dot{m}_{wt} Cp_{wt}(T_5 - T_4)]}{dCp_{wt}}\Delta Cp_{wt}\right\}^2 + \\ &\left\{\frac{d[\dot{m}_{wt} Cp_{wt}(T_5 - T_4)]}{dT_5}\Delta T_5\right\}^2 + \left\{\frac{d[\dot{m}_{wt} Cp_{wt}(T_5 - T_4)]}{dT_4}\Delta T_4\right\}^2\end{aligned}} \tag{E.8}$$

$$\Delta Q_{ehe} = \sqrt{\begin{aligned}&[Cp_{wt}(T_5 - T_4)\Delta \dot{m}_{wt}]^2 + [\dot{m}_{wt}(T_5 - T_4)\Delta Cp_{wt}]^2 + \\ &(\dot{m}_{wt} Cp_{wt}\Delta T_5)^2 + (-\dot{m}_{wt} Cp_{wt}\Delta T_4)^2\end{aligned}} \tag{E.9}$$

E.2.4 Thermal CHP Efficiency

From: $\eta_{th_{chp}} = \frac{Q_{ehe}}{Q_f}$ *(E.10)*

$$\Delta \eta_{th_{chp}} = \sqrt{\left[\frac{d(Q_{ehe}/Q_f)}{dQ_f}\Delta Q_f\right]^2 + \left[\frac{d(Q_{ehe}/Q_f)}{dQ_{ehe}}\Delta Q_{ehe}\right]^2} \tag{E.11}$$

$$\Delta \eta_{th_{chp}} = \sqrt{\left(\frac{-Q_{ehe}}{Q_f^{\,2}}\Delta Q_f\right)^2 + \left(\frac{1}{Q_f}\Delta Q_{ehe}\right)^2} \tag{E.12}$$

E.2.5 Total CHP Efficiency

From: $\eta_{chp} = \dfrac{w_e + Q_{ehe}}{Q_f}$ (E.13)

$$\Delta\eta_{chp} = \sqrt{\left\{\frac{d\left[\left(w_e + Q_{ehe}\right)/Q_f\right]}{dQ_f}\Delta Q_f\right\}^2 + \left\{\frac{d\left[\left(w_e + Q_{ehe}\right)/Q_f\right]}{dw_e}\Delta w_e\right\}^2 + \left\{\frac{d\left[\left(w_e + Q_{ehe}\right)/Q_f\right]}{dQ_{ehe}}\Delta Q_{ehe}\right\}^2} \quad (E.14)$$

$$\Delta\eta_{chp} = \sqrt{\left[\frac{-\left(w_e + Q_{ehe}\right)}{Q_f^{\,2}}\Delta Q_f\right]^2 + \left(\frac{1}{Q_f}\Delta w_e\right)^2 + \left(\frac{1}{Q_f}\Delta Q_{ehe}\right)^2} \quad (E.15)$$

E.2.6 Heat Pump Power Input

$w_{hp} = I_{hp}V_{hp}$ (E.16)

Note V_{hp} represent the RMS voltage

$$\Delta w_{hp} = \sqrt{\left[\frac{d\left(I_{hp}V_{hp}\right)}{dI_{hp}}\Delta I_{hp}\right]^2 + \left[\frac{d\left(I_{hp}V_{hp}\right)}{dV_{hp}}\Delta V_{hp}\right]^2} \quad (E.17)$$

$$\Delta w_{hp} = \sqrt{(V_{hp}\Delta I_{hp})^2 + (I_{hp}\Delta V_{hp})^2} \quad (E.18)$$

E.2.7 Heat Pump Thermal Delivery

From: $Q_{hp} = \dot{m}_{wt} Cp_{wt} (T_3 - T_2)$ *(E.19)*

$$\Delta Q_{hp} = \sqrt{\begin{array}{l} \left\{\dfrac{d[\dot{m}_{wt} Cp_{wt} (T_3 - T_2)]}{d\dot{m}_{wt}} \Delta\dot{m}_{wt}\right\}^2 + \left\{\dfrac{d[\dot{m}_{wt} Cp_{wt} (T_3 - T_2)]}{dCp_{wt}} \Delta Cp_{wt}\right\}^2 + \\ \left\{\dfrac{d[\dot{m}_{wt} Cp_{wt} (T_3 - T_2)]}{dT_3} \Delta T_3\right\}^2 + \left\{\dfrac{d[\dot{m}_{wt} Cp_{wt} (T_3 - T_2)]}{dT_2} \Delta T_2\right\}^2 \end{array}} \qquad (E.20)$$

$$\Delta Q_{hp} = \sqrt{\begin{array}{l} [Cp_{wt} (T_3 - T_2) \Delta\dot{m}_{wt}]^2 + [\dot{m}_{wt} (T_3 - T_2) \Delta Cp_{wt}]^2 + \\ (\dot{m}_{wt} Cp_{wt} \Delta T_3)^2 + (-\dot{m}_{wt} Cp_{wt} \Delta T_2)^2 \end{array}} \qquad (E.21)$$

E.2.8 Heat Pump COP

From: $COP = \dfrac{Q_{hp}}{w_{hp}}$ *(E.22)*

$$\Delta COP = \sqrt{\left[\frac{d(Q_{hp}/w_{hp})}{dw_{hp}} \Delta w_{hp}\right]^2 + \left[\frac{d(Q_{hp}/w_{hp})}{dQ_{hp}} \Delta Q_{hp}\right]^2} \qquad (E.23)$$

$$\Delta COP = \sqrt{\left(\frac{-Q_{hp}}{w_{hp}^{\ 2}} \Delta w_{hp}\right)^2 + \left(\frac{1}{w_{hp}} \Delta Q_{hp}\right)^2} \qquad (E.24)$$

E.2.9 CHP/HP Thermal Efficiency

From: $\eta_{th_{chphp}} = \dfrac{Q_{ehe} + Q_{ehe}}{Q_f}$ (E.25)

$$\Delta\eta_{th_{chphp}} = \sqrt{\begin{aligned}&\left\{\frac{d\left[\left(Q_{hp}+Q_{ehe}\right)/Q_f\right]}{dQ_f}\Delta Q_f\right\}^2 + \left\{\frac{d\left[\left(Q_{hp}+Q_{ehe}\right)/Q_f\right]}{dQ_{hp}}\Delta Q_{hp}\right\}^2 \\ &+ \left\{\frac{d\left[\left(Q_{hp}+Q_{ehe}\right)/Q_f\right]}{dQ_{ehe}}\Delta Q_{ehe}\right\}^2\end{aligned}} \quad (E.26)$$

$$\Delta\eta_{th_{chphp}} = \sqrt{\left[\frac{-\left(Q_{hp}+Q_{ehe}\right)}{Q_f^{\,2}}\Delta Q_f\right]^2 + \left(\frac{1}{Q_f}\Delta Q_{hp}\right)^2 + \left(\frac{1}{Q_f}\Delta Q_{ehe}\right)^2} \quad (E.27)$$

E.2.10 Total CHP/HP Efficiency

From: $\eta_{chphp} = \dfrac{w_e + Q_{ehe} + Q_{hp}}{Q_f}$ (E.28)

$$\Delta\eta_{chphp} = \sqrt{\begin{aligned}&\left[\frac{d\left(w_e + Q_{ehe} + Q_{hp}/Q_f\right)}{dQ_f}\Delta Q_f\right]^2 + \left[\frac{d\left(w_e + Q_{ehe} + Q_{hp}/Q_f\right)}{dw_e}\Delta w_e\right]^2 + \\ &\left[\frac{d\left(w_e + Q_{ehe} + Q_{hp}/Q_f\right)}{dQ_{ehe}}\Delta Q_{ehe}\right]^2 + \left[\frac{d\left(w_e + Q_{ehe} + Q_{hp}/Q_f\right)}{dQ_{hp}}\Delta Q_{hp}\right]^2\end{aligned}}$$

(E.29)

$$\Delta\eta_{chphp} = \sqrt{\left[\frac{\left(w_e + Q_{ehe} + Q_{hp}\right)}{Q_f^{\,2}}\Delta Q_f\right]^2 + \left(\frac{1}{Q_f}\Delta Q_{ehe}\right)^2 + \left(\frac{1}{Q_f}\Delta Q_{hp}\right)^2 + \left(\frac{1}{Q_f}\Delta w_e\right)^2}$$

(E.30)

E.2.11 Total Plant Thermal Output (Q_{th})

From: $Q_{th} = Q_{ehe} + Q_{hp}$ (E.31)

$$\Delta Q_{th} = \sqrt{\left[\frac{d(Q_{ehe} + Q_{hp})}{dQ_{ehe}} \Delta Q_{ehe}\right]^2 + \left[\frac{d(Q_{ehe} + Q_{hp})}{dQ_{hp}} \Delta Q_{hp}\right]^2} \quad (E.32)$$

$$\Delta Q_{th} = \sqrt{\Delta Q_{ehe}{}^2 + \Delta Q_{hp}{}^2} \quad (E.33)$$

E.2.12 Adjusted Power Delivery (w_e')

From: $w_e' = w_e - w_{hp}$ (E.34)

$$\Delta w_e' = \sqrt{\left[\frac{d(w_e - w_{hp})}{dw_e} \Delta w_e\right]^2 + \left[\frac{d(w_e - w_{hp})}{dw_{hp}} \Delta w_{hp}\right]^2} \quad (E.35)$$

$$\Delta w_e' = \sqrt{\Delta w_e{}^2 - \Delta w_{hp}{}^2} \quad (E.36)$$

E.3.1. CHP Mode Sensitivity Analysis (see section 6.3.1.1)

I_{gen}	ΔI_{gen}	V_{gen}	ΔV_{gen}	w_e	Δw_e
A	A	V	V	kW	kW
4.180	0.0122	234.410	0.0217	0.980	0.003

E.3

m_{wt}	Δm_{wt}	Cp	ΔCp	T_5	ΔT_5	T_4	ΔT_4	Q_{ehe}	ΔQ_{ehe}
kg/s	kg/s	kJ/kgC	kJ/kgC	c	c	c	c	kW	kW
0.084	0.0108	4.200	0.000	46.550	0.0204	38.350	0.0135	2.893	0.373

E.9

Q_f	ΔQ_f	η_e	$\Delta\eta_e$
kW	kW		
6.5300	0.0000	0.1501	0.0004

E.6

Q_f	ΔQ_f	$\eta_{th\text{-}chp}$	$\Delta_{hth\text{-}chp}$
kW	kW		
6.530	0.000	0.443	0.057

E.12

Q_f	ΔQ_f	η_{chp}	$\Delta\eta_{chp}$
kW	kW		
6.530	0.000	0.593	0.057

E.15

E.3.2. HP Mode Sensitivity Analysis (see section 6.3.1.2)

I_{gen}	ΔI_{gen}	V_{gen}	ΔV_{gen}	w_e	Δw_e
A	A	V	V	kW	kW
3.751	0.0122	239.900	0.0217	0.900	0.003

E.3

m_{wt}	Δm_{wt}	Cp	ΔCp	T_5	ΔT_5	T_4	ΔT_4	Q_{ehe}	ΔQ_{ehe}
kg/s	kg/s	kJ/kgC	kJ/kgC	c	c	c	c	kW	kW
0.061	0.0108	4.200	0.000	60.170	0.0204	50.330	0.0135	2.521	0.448

E.9

m_{wt}	Δm_{wt}	Cp	ΔCp	T_3	ΔT_3	T_2	ΔT_2	Q_{hp}	ΔQ_{hp}
kg/s	kg/s	kJ/kgC	kJ/kgC	c	c	c	c	kW	kW
0.061	0.0108	4.200	0.000	46.670	0.0119	37.400	0.0168	2.375	0.422

E.21

Q_f	ΔQ_f	η_e	$\Delta \eta_e$
kW	kW		
7.0000	0.0000	0.1286	0.0004

E.6

Q_f	ΔQ_f	$\eta_{th\text{-}chp}$	$\Delta_{hth\text{-}chp}$
kW	kW		
7.000	0.000	0.360	0.064

E.12

Q_f	ΔQ_f	η_{chp}	$\Delta \eta_{chp}$
kW	kW		
7.000	0.000	0.489	0.064

E.15

Q_{th}	ΔQ_{th}
kW	kW
4.896	0.616

E.33

I_{hp}	ΔI_{hp}	V_{hp}	ΔV_{hp}	w_{hp}	Δw_{hp}
A	A	V	V	kW	kW
3.750	0.004	239.390	0.217	0.898	0.001

E.18

COP	ΔCOP
2.646	0.470

E.24

$\eta_{th\text{-}chp/hp}$	$\Delta\eta_{th\text{-}chp/hp}$
0.699	0.088

E.27

w_e'	$\Delta w_e'$
kW	kW
0.002	0.003

E.36

$\eta_{chp/hp}$	$\Delta\eta_{chp/hp}$
0.700	0.088

E.30

E.3.3. CHP/HP Sensivity Analysis (See section 6.3.1.3)

I_{gen}	ΔI_{gen}	V_{gen}	ΔV_{gen}	w_e	Δw_e
A	A	V	V	kW	kW
4.580	0.0122	248.920	0.0217	1.140	0.003

E.3

m_{wt}	Δm_{wt}	Cp	ΔCp	T_5	ΔT_5	T_4	ΔT_4	Q_{ehe}	ΔQ_{ehe}
kg/s	kg/s	kJ/kgC	kJ/kgC	c	c	c	c	kW	kW
0.060	0.0108	4.200	0.000	53.170	0.0204	44.120	0.0135	2.281	0.412

E.9

m_{wt}	Δm_{wt}	Cp	ΔCp	T_3	ΔT_3	T_2	ΔT_2	Q_{hp}	ΔQ_{hp}
kg/s	kg/s	kJ/kgC	kJ/kgC	c	c	c	c	kW	kW
0.060	0.0108	4.200	0.000	41.220	0.0119	31.860	0.0168	2.359	0.426

E.19

Q_f	ΔQ_f	η_e	$\Delta\eta_e$
kW	kW		
6.6500	0.0000	0.1714	0.0005

E.6

Q_f	ΔQ_f	$\eta_{th\text{-}chp}$	$\Delta_{hth\text{-}chp}$
kW	kW		
6.650	0.000	0.343	0.062

E.12

Q_f	ΔQ_f	η_{chp}	$\Delta\eta_{chp}$
kW	kW		
6.650	0.000	0.514	0.062

E.15

Q_{th}	ΔQ_{th}
kW	kW
4.639	0.593

E.33

I_{hp}	$\Delta\iota\eta\pi$	V_{hp}	ΔV_{hp}	w_{hp}	Δw_{hp}
A	A	V	V	kW	kW
3.690	0.004	248.960	0.217	0.919	0.001

E.18

COP	ΔCOP
2.568	0.464

E.24

$\eta_{th\text{-}chp/hp}$	$\Delta\eta_{th\text{-}chp/hp}$
0.698	0.089

E.27

w_e'	$\Delta w_e'$
kW	kW
0.221	0.003

E.36

$\eta_{chp/hp}$	$\Delta\eta_{chp/hp}$
0.731	0.089

E.30

Appendix F Model Codes

This appendix will reconcile the nomenclature used in the thesis and that required by the programming language (MS Visual Basic 4), prior to the presentation of sub-routine codes.

F.1 Nomenclature Translation

Owing to the restriction of code script the nomenclature employed in the *Concept Evaluation Model* code differs from the nomenclature used in the analysis model. Derivation table F.1 reconciles the two nomenclatures.

Table F.1. Nomenclature Translation

Thesis Nomenclature	Code Nomenclature	Description
D_e	De	Electrical demand.
D_{th}	Dth	Thermal Demand.
D_{ea}	Dea	Adjusted electrical demand.
D_{tha}	Dtha	Adjusted thermal demand
D_{thap}	Dthap	Adjusted potential thermal demand.
Lf	Lf	Load factor.
w_e	We	Plant electrical output.
w_s	Ws	Surplus generator capacity.
T_n	T(n)	Temperature.
$w_e - D_e$	Wd	Delivered power.
Q_{hp}	Qthhp	Heat pump thermal delivery.
Q_{hpp}	Qthhpp	Heat pump potential thermal delivery.
Q_{ehe}	Qthehe	EHE thermal delivery.
Q_{ehep}	Qthehep	EHE potential thermal delivery.
Q_{th}	Qth	Total thermal output.
Q_{thp}	Qthp	Total potential thermal output.
Q_f	Qf	Fuel input.

T_{bulk}	Tblk	Heat exchanger bulk water temperature.
$Q_{ehe\text{-}in}$	Qehein	Steady state EHE heat input.
V_{ehe}	Vehe	EHE volume.
m_{wt}	Mw	LPW system water flow.
Cp_{wt}	Cpw	CP of water.
τ	Tua	Time step size.
Q_{hp-in}	Qthhp-in	Steady state heat pump heat exchanger input.
η_e	He	Engine/generator set electrical efficiency.
λ_m	Ml	Engine thermal output constant.
λ_c	Cl	Engine thermal output constant.
A_{η_e} to D_{η_e}	Ah to Dh	Engine/generator set efficiency constants.
COP_{max}	Mcop	Heat Pump COP constant.
COP_{min}	Ccop	Heat Pump COP constant.
K	Lcop	Part load COP exponential constant.
A_{cop2} to D_{cop2}	Ahp to Dhp	By pass heat pump constants.
U_f	Ucf	Unit fuel cost.
U_e	UcE	Unit electricity cost.
U_m	UcM	Unit maintenance cost.
C_r	Cr	Running costs.
Savings	Savings	
$Savings_p$	Savingsp	Potential steady state savings.
η_{blr}	Hblr	Boiler efficiency.
$S_{coal}, S_{ng}, S_{oil}$	Scoal, Sng, Soil	Specific emissions constants.
$\eta_{coal}, \eta_{ccgct}, \eta_{oil}, \eta_{grid}$	hcoal, hccgt, hoil, htrans	Utility supply efficiencies.
m_{co2}	Mco2	Mass CO_2 evolved.

F.2 *run_loop* Code (Reference Section 7.3)

```
Sub run_loop()
IDC = 0
'Set datasetout = datout.OpenRecordset(namedatout)
datsetin.MoveLast
tuamax = datsetin.Fields("ID")
res = datheaderin.Fields("res")
datsetin.MoveFirst
datsetin.MoveFirst
resc = datsetin.Fields("res")
   Do
   datsetin.MoveNext
   resc = resc + datsetin.Fields("res")
   Loop Until datsetin.Fields("ID") = tuamax
   nc = 0
datsetin.MoveFirst
For n = 1 To tuamax
De = datsetin.Fields("De")
   Form1.txtDe = "De =" + Format((De), "0.00") + "kW"
   Form1.txtDe.Refresh
   Dth = datsetin.Fields("Dth")
   Form1.txtDth = "Dth =" + Format((Dth), "0.00") + "kW"
   Form1.txtDth.Refresh
   mw = datsetin.Fields("mw")
   T(2) = datsetin.Fields("T2") + 273
   T(4) = datsetin.Fields("T4") + 273
   ws = datsetin.Fields("ws")
   res = datsetin.Fields("res")

      'Heat pump cold start adjustment
       If n <= 35 Then
         Fthcap = 0.014857 * (35 - n)
         Vehe = 8 + (4 * (0.11428 * (35 - n)))
         End If

            For nsub = 1 To res Step 15 'tua
             tt = tt + 1

             'analytical subs

             T(2) = T(1)
             nc = nc + (1 * tua)
             nc2 = nc2 + (1 * tua)

             'mode filter

      If Form1.option1.Value = 1 Then chp_calc
      If Form1.option2.Value = 1 Then chphp_calc
      If Form1.option3.vaule = 1 Then hp_calc
      heat_load
      'displays
      Dec1 = Dec1 + (De * tua)
      Dthc1 = Dthc1 + (Dth * tua)
      wec1 = wec1 + (we * tua)
      wdc1 = wdc1 + (wd * tua)
      Qthc1 = Qthc1 + (Qth * tua)
      LFc1 = LFc1 + (LF * tua)
      QFc1 = QFc1 + (Qf * tua)
```

```
Qthehec1 = Qthehec1 + (Qthehe * tua)
Qthhpc1 = Qthhpc1 + (Qthhp * tua)
LFc2 = LFc2 + (LF * tua)
Dec2 = Dec2 + (De * tua)
Dthc2 = Dthc2 + (Dth * tua)
wec2 = wec2 + (we * tua)
wdc2 = wdc2 + (wd * tua)
Qthc2 = Qthc2 + (Qth * tua)
Qthehec2 = Qthehec2 + (Qthehe * tua)
Qthhpc2 = Qthhpc2 + (Qthhp * tua)
Qfc2 = Qfc2 + (Qf * tua)
wdc2 = wec2 - wdc2
   For tr = 2 To 5
   Tc2(tr) = Tc2(tr) + T(tr)
   Next tr
   If nc2 = res Then
   DeAv1 = Dec2 / (res)
  DthAv1 = Dthc2 / (res)
  weAv1 = wec2 / (res)
  QthAv1 = Qthc2 / (res)
  QtheheAv1 = Qthehec2 / (res)
  QthhpAv1 = Qthhpc2 / (res)
  wdAv1 = wdc2 / (res)
  QfAv1 = Qfc2 / (res)
  wsAv1 = wdc2 / (res)
  LFav1 = LFc2 / res
    For tr = 2 To 5
     TAv(tr) = Tc2(tr) / (res / tua)
     Next tr
   nc2 = 0
   Dec2 = 0
   Dthc2 = 0
   wec2 = 0
   wdc2 = 0
   Qthc2 = 0
   Qthehec2 = 0
   Qthhpc2 = 0
   wsc2 = 0
   Qfc2 = 0
IDC = IDC + 1
'dataout insertion
datasetout.AddNew
datasetout.Fields("ID") = IDC
datasetout.Fields("Timem") = tua * tt / 86400
datasetout.Fields("Tm1") = TAv(1)
datasetout.Fields("Tm2") = TAv(2)
datasetout.Fields("Tm3") = TAv(3)
datasetout.Fields("Tm4") = ucE
datasetout.Fields("Tm5") = weAv1 / 1 * 100
datasetout.Fields("Qehe") = QtheheAv1
datasetout.Fields("Qhp") = QthhpAv1
              If Dth > 0 Then
              datasetout.Fields("Qtot") = QthAv1
              Else
               datasetout.Fields("Qtot") = 0
              End If

                datasetout.Fields("Qin") = QfAv1
                        If De > 0 Then
```

```
                    datasetout.Fields("w") = weAv1
                     Else
                     datasetout.Fields("w") = 0
                       End If
            datasetout.Fields("ps") = wsAv1
            datasetout.Fields("Lcop") = Lcop
            datasetout.Update
            End If
      Next nsub
      datsetin.MoveNext
Next n
wdc1 = 0
wec1 = 0
Dec1 = 0
Qthc1 = 0
Dthc1 = 0
QFc1 = 0
End Sub
```

F.3 *data_head* Code (Reference: Section 7.3)

```
Sub datheader_load()

Set datin = OpenDatabase("indat")
Set datheaderin = datin.OpenRecordset("SELECT * FROM indexset ")
datheaderin.MoveLast
nt = datheaderin.Fields("ID")
datheaderin.MoveFirst
For n = 1 To nt
  Form1.Combo1.AddItem datheaderin.Fields("datset")
  datheaderin.MoveNext
  Next n
End Sub
```

F.4 *chp_calc* Code (Reference: Section 7.4.1)

```
Sub chp_calc()
T(3) = T(2)
T(4) = T(3)
If De < wmax Then LF = De / wmax
If De >= wmax Then LF = 1
we = wmax * LF
Qthehep = (ml * we) + Cl
If Qthehep > Dth Then
   we = (Dth / ml) - Cl
   If we < 0 Then we = 0
   LF = we / wmax
   Qthehep = (ml * we) + Cl
   End If
ws = we - De
wd = we
ws = 0
Qthhp_calc
'T(4) = datsetin.Fields("T4") + 273 : reserved for validation error trap
Qthehe_calc
Qth = Qthhp + Qthehe
h_calc
Dea = De - we
Dtha = Dth - Qth
Qthhpp = 0
Qthp = Qthhpp + Qthehep
Dthap = Dth - Qthp
economic_calc
If savingsp < 0 Then
   LF = 0
   we = 0
   ws = 0
   wd = 0
   Qf = 0
   Dea = De
   T(2) = Told(2)
   T(3) = Told(3)
   T(4) = Told(4)
   T(5) = Told(5)
   'T(4) = datsetin.Fields("T4") + 273 :Ditto
   Qthhp_calc
   Qthehe_calc
   Qth = Qthhp + Qthehe
   Dtha = Dth - Qth
   End If
End Sub
```

F.5 *chphp_calc* Code (Reference : Section 7.4.2)

```
Sub chphp_calc()
If De >= wmax Then
   chp_calc
   Else
   LF = 1
   we = wmax
   ws = we - De
   wd = we - ws
   cop_calc
   Qthhpp = cop * ws
   Qthehep = (ml * we) + Cl
   Qthp = Qthhpp + Qthehep
   If Dth <= 0 Then
      LF = 0:ws = 0:we = 0:wd = 0:Qthehep = 0:Qhpthp = 0
      Else
      If Qthp > Dth And LF > 0 Then
         Do
         LF = LF - 0.01
         we = LF * wmax
         ws = we - De
         cop_calc
         Qthhpp = cop * ws
         Qthehep = (ml * we) + Cl
            If LF <= 0 Then
            LF =0:Qthehep = 0:Qhpthp =0:we = 0:wd = 0
            End If
         Qthp = Qthhpp + Qthehep
         Loop Until Qthp <= Dth
         End If
      End If
   cop_calc
   Qthhp_calc
   Qthehe_calc
   Qth = Qthhp + Qthehe
   h_calc
   Dea = De - wd
   Dtha = Dth - Qth
   Dthap = Dthp - Qthp
   economic_calc
   If savingsp <= 0 Then
      chp2_calc
      End If
   If savingsp < 0 Then
      LF = 0:we = 0:Qf = 0: ws = 0:wd = 0: Dea = De
      T(2) = Told(2)
      T(3) = Told(3)
      T(4) = Told(4)
      T(5) = Told(5)
      Qthhp_calc
      Qthehe_calc
      Qth = Qthhp + Qthehe
      Dtha = Dth - Qth
      End If
   End If
End Sub
```

F.6 *Qthehe_calc* Code (Reference : Section 7.4.3)

```
Sub Qthehe_calc()
Qthehein = (ml * we) + Cl
If we <= 0 Then
   Qthehein = 0
   End If

Told(4) = T(4)
Told(5) = T(5)

Tblk = (((Vehe - (mw * tua)) * T(5)) + ((mw * tua) * T(4))) / Vehe
T(5) = ((Qthehein * tua) / (Vehe * Cpw)) + Tblk
Qthehe = (mw * Cpw * (T(5) - T(4)))
End Sub
```

F.7 *Qthhp_calc* Code (Reference : Section 7.4.4)

```
Sub Qthhp_calc()
Qthhpin = ws * cop
Told(2) = T(2)
Told(3) = T(3)
Tblkhp = (((Vhp - (mw * tua)) * T(3)) + ((mw * tua) * T(2))) / Vhp
T(3) = ((Qthhpin * tua) / (Vhp * Cpw)) + Tblkhp
  Qthhp = (mw * Cpw * (T(3) - T(2)))
End Sub
```

F.8 *h_calc* Code (Reference : Section 7.4.5)

```
Sub h_calc()
he = ((Ah * (we ^ 3)) + (Bh * (we ^ 2)) + (Ch * we) - Dh) / 100
Qf = we / he
End Sub
```

F.9 *cop_calc* Code (Reference Section 7.4.6)

```
Sub cop_calc()
' Option 1 Experimental
cop = (2.53 * whp) + 0.234
'Option 2 By-Pass
If we > 0 Then
   cop = (Ahp * (ws ^ 3)) + (Bhp * (ws ^ 2)) + (Chp * ws) + Dhp
   Else
   cop = 0
   wd = 0
   End If
'Option 3 Gernalised Exponential Function
If ws <= 0 Then
   cop = 0
   Else
   cop = ((1 - (Exp(ws * Lcop))) * Mcop) + Ccop
   End If
End Sub
```

F.10 *ecomonic_calc* Code (Reference : section 7.4.7)

```
Sub economic_calc()

Cr = (Qf * ucf) + (we * ucM)
Cad = (Dea * ucE) + (Dtha / hblr * ucf)
savings = ((wd * ucE) + (Qth * ucf / hblr)) - Cr
savingsp = ((wd * ucE) + (Qthp * ucf / hblr)) - Cr

End Sub
```

F.11 *enviro_calc* Code (Reference : Section 7.4.8)

```
Sub enviro_calc()

Sgrid = ((Scoal * Pcoal) / (hcoal)) + ((Soil * Poil) / (hoil)) + ((Sng * Pccgt) / (hccgt))
mco2c = (Dth / hblr * Sng) + (Sgrid * De / htrans)
mco2 = (Dtha / hblr * Sng) + (Sgrid * Dea / htrans) + (Qf * Sng)
End Sub
```